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MULTIPLE EXPANSION REFRIGERATION CYCLES

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SUPERVISOR

Professor J Gryzagoridis

I, John Arthur Browne submit this thesis in partial fulfilment of the requirements for the degree of Master of Science in Engineering. I claim that this is my original work and that it has not been submitted in this or any similar form for any degree at any University.

Signed

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September 1986

ABSTRACT

The work described in this thesis deals with some theoretical and practical aspects of multiple expansion vapour compression refrigeration cycles.

Research on refrigeration cycles has concentrated on the development of the single and two stage vapour compression cycle, the absorption refrigeration cycle and the development and evaluation of new refrigerants and absorbents. This has occured to the extent that further improvements in the performance of these cycles that can be economically implemented for small scale units are unlikely to be found.

This work undertook to carry out the following :

i) A theoretical study of the multiple expansion vapour compression cycle aimed at determining the theoretical performance of the cycle and comparing it to that of other commonly used refrigeration cycles.

ii) An experimental study of a hybrid refrigeration cycle which combined a multiple expansion vapour compression cycle with an absorption cycle. This study aimed to obtain an indication of the performance and practical usefulness of the cycle and to prove that hybrid cycles can be operated successfully. The cycle used dimethyl ether - tetra ethylene glycol as the absorbent and R22 as the refrigerant.

The results obtained indicated that signifigant improvements in performance are theoretically possible with multiple expansion vapour compression cycles. However considerable development of a suitable means to achieve the multiple compressions required for the cycle is required.

The experimental work with the hybrid cycle showed that multiple refrigerant expansions were possible in practice. This successfully achieved the first step in the production of a true multiple expansion vapour compression cycle as studied theoretically.

Finally it was found that the multiple expansion - absorption cycle works satisfactorily in practice but its low performance and high cost limit its usefulness to special applications.

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CHAPTER 1

INTRODUCTION

1.1 A BRIEF HISTORY OF REFRIGERATION

The earliest form of refrigeration known was the use of natural ice to chill and preserve food [1,2]. Blocks of ice were obtained by cutting up the iced surfaces of frozen lakes and ponds. These blocks were stored and transported in insulated containers and used to refrigerate food and other items by placing them in close contact with the ice. As the ice melted it absorbed latent heat as well as sensible heat from the surroundings thereby producing a refrigerating effect.

The first successful preparation of ice by artificial means was achieved by Jacob Perkins in 1834 [2]. His apparatus was the forerunner of the modern vapour compression refrigeration system and consisted of a hand operated compressor, a water cooled condenser and an evaporator submerged in a liquid bath [1]. Ether was used as the refrigerant in this apparatus.

The principle of the absorption refrigeration system was discovered by Faraday in 1824 [2] but it was not until 1855 that the first refrigerating apparatus based on these absorption principles was produced.

During the nineteenth century there were many pioneers in refrigeration and many devices were proposed and tried. Among them was the first compressed air system which was patented by Dr. John Gorrie in 1851 [1].

At the end of the nineteenth century refrigeration systems were characterised by bulky, heavy and generally "over engineered" units. Compressors were still driven by steam, operating speeds were well below 50 rpm and the use of refrigeration was restricted to industrial applications.

Domestic refrigeration first made its appearance in the early nineteen hundreds and the first "Kelvinator" was available on the American market in 1918 [2]. From this time onwards commercial, industrial and domestic refrigerating systems began to really take hold and sales increased continually. New applications for refrigeration were found in fields such as medicine, food processing and air conditioning and the market for refrigeration systems expanded rapidly. With the increasing volume of sales, many new companies specialising in the supply of refrigeration equipment appeared and the various systems were quickly improved.

Today refrigeration has become an indispensable part of our lifestyle and is essential in the manufacture and preservation of food, medicine and other essential commodities for millions of city dwellers. Due to the present advanced state of refrigeration technology further improvements in the performance of refrigerating systems, that could be economically implemented for small scale refrigerating units, are unlikely to be sucessfully developed. However, as a result of the present high and ever increasing cost of energy, small improvements in the performance of large industrial and commercial refrigeration systems could lead to significant financial savings. Thus any possibilities which exist for the improvement of the refrigeration systems used in commerce and industry are worth a close investigation.

1.2 BASIC REFRIGERATION CONCEPTS

Present day refrigeration systems generally use one of two well known and established cycles. These are the vapour compression (V.C.) and the vapour absorption (V.A.) cycles. Both of these cycles achieve the refrigerating effect by the same means i.e. the evaporation of a liquid refrigerant at low pressure and temperature.

When a liquid is introduced into an evacuated vessel some of the liquid evaporates until an equilibrium is established and the pressure in the vessel equals the vapour pressure of the liquid at that temperature. At this equilibrium there is no nett evaporation of the refrigerant and the vapour continually condenses at the same rate at which the liquid evaporates. The latent heat absorbed by the evaporating molecules exactly balances the latent heat released by the condensing molecules and there is no nett heat transfer. When some of the vapour is removed from the vessel the pressure is initially reduced to some pressure below the vapour pressure of the liquid. The equilibrium which existed is disturbed and in order to raise the pressure and restore the equilibrium more molecules evaporate than condense. The latent heat absorbed by the evaporating molecules excedes that released by the condensing molecules and there is a nett flow of heat into the vapour. This heat is supplied from the sensible heat of the refrigerant and the temperature falls [3].

This mechanism is used to achieve the refrigerating effect in both the vapour compression and the vapour absorption refrigeration cycles. Some suitable liquid refrigerant is maintained at a low pressure with vapour being continuously removed from the vapour space above it. The latent heat which is absorbed by the refrigerant as it evaporates at low pressure and temperature produces the refrigerating effect.

In order for a small quantity of the refrigerant to be re-used in a closed cycle, the refrigerant evaporated to produce the refrigerating effect must be returned to the liquid state. To enable the condensation of the refrigerant vapour, the pressure is raised until the saturation temperature is higher than the temperature of some convenient heat sink. This enables the refrigerant vapour to reject the latent heat of evaporation to the heat sink and the vapour is able to condense to liquid. This heat rejection is done in a condenser which is usually cooled by water or air at ambient temperature.

The vapour compression and absorption cycles differ in the method used to transport the refrigerant from the low pressure side (evaporator) to the high pressure side (condenser) of the system. The V.C. cycle uses a compressor while the V.A. cycle employs a more complicated system in which the refrigerant vapour is absorbed into a carrier liquid on the low pressure side and released again on the high pressure side.

1.3 THE VAPOUR COMPRESSION REFRIGERATION CYCLE

The vapour compression cycle is the simplest refrigeration cycle and consists of an evaporator, a condenser, a compressor and an expansion device. Figure 1.1 shows the basic V.C. cycle.

The V.C. cycle can be represented on a pressure enthalpy diagram as shown in Figure 1.2. Refering to this diagram, saturated vapour at condition (1) is compressed (assumed isentropically) to condition (2) which is superheated vapour at the condenser pressure. This high temperature vapour passes into the condenser where it is first de-superheated and then condensed. The resulting saturated liquid at condition (3) is then expanded with constant enthalpy to condition (4). During the expansion flash gas (saturated vapour) at condition (1) and saturated liquid at condition (5) is formed with condition (4) being an equilibrium mixture of these two components. The saturated liquid at condition (5) evaporates in the evaporator to form saturated vapour at condition (1) and in the process absorbs its latent heat of evaporation and produces a refrigerating effect. This vapour along with the flash gas from the expansion is then compresssed to the condenser pressure thereby completing the cycle.

The coefficient of performance of the V.C. cycle has been defined as :

$$C.O.P = R.E. / W \qquad \text{---} \quad (1.1)$$

Where

C.O.P = Coefficient of performance of the vapour compression cycle.
R.E. = Refrigerating effect (kJ/kg).
W = Compressor work (kJ/kg).

In most cases the refrigerating effect achieved is greater than the work input to the cycle and the coefficient of performance is greater than unity. From equation (1.1) it can be seen that in order to increase the coefficient of performance the compressor work must be decreased and or the refrigerating effect increased.

A more detailed description of the single stage V.C. refrigerating cycle can be found in Appendix A as well as references 3,4,5 and 6.

FIG 1.1 SINGLE STAGE VAPOUR COMPRESSION REFRIGERATION CYCLE

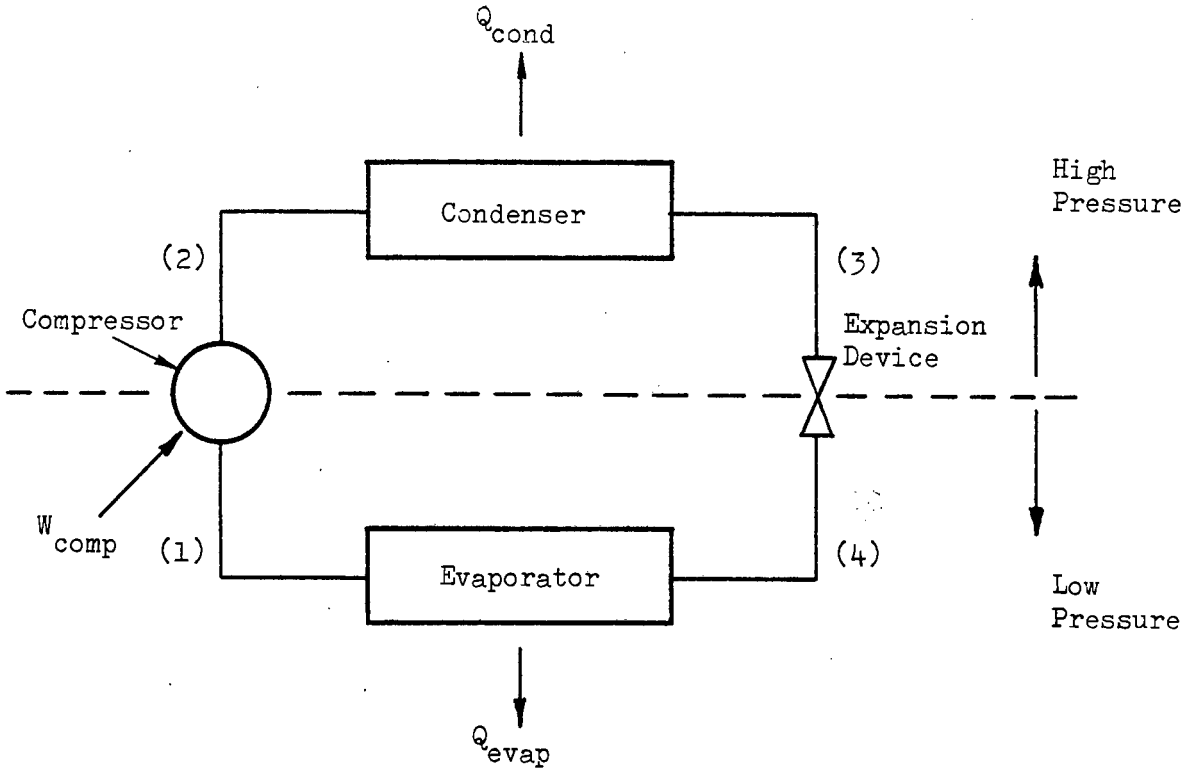
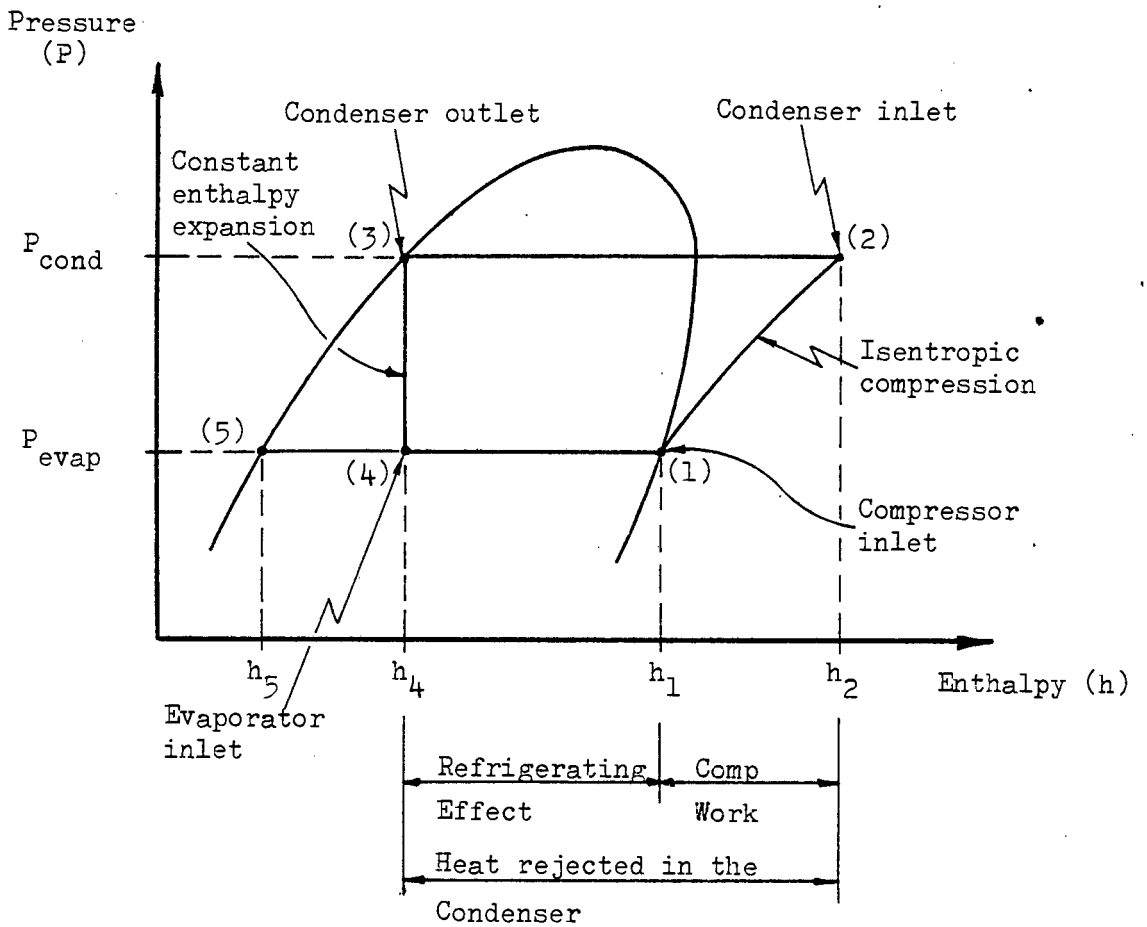


FIG 1.2 SINGLE STAGE VAPOUR COMPRESSION CYCLE ON A PRESSURE - ENTHALPY DIAGRAM



1.4 THE VAPOUR ABSORPTION REFRIGERATION CYCLE

The vapour absorption refrigeration cycle is more complicated than the vapour compression cycle and consists of an evaporator, a condenser, an absorber, a generator, a liquid pump and two expansion devices. Figure 1.3 shows the basic V.A. cycle.

Instead of the compressor used in the V.C. cycle the V.A. cycle uses a circulating carrier fluid (the absorbent) to transport the refrigerant vapour from the evaporator to the condenser. Refering to Figure 1.3, refrigerant vapour from the evaporator is passed to the absorber where it is absorbed into the absorbent. The resulting solution, rich in absorbed refrigerant, is pumped to the generator using a small liquid pump. In the generator this refrigerant laden absorbent (strong solution) is heated. As the temperature is increased the solubility of the refrigerant in the absorbent decreases and some of the refrigerant is released. The liberated refrigerant vapour passes to the condenser where it is cooled and condensed. The resulting liquid refrigerant is then expanded to the evaporator pressure in the usual way thereby completing the refrigerant path. The absorbent in the generator, now lean in absorbed refrigerant (weak solution), is returned to the absorber where it absorbs some more refrigerant vapour thereby completing the absorbent path.

The mechanical work which must be supplied to the pump is usually negligibly small as a result of the low specific volume of the liquid absorbent. However considerable quantities of low grade heat must be supplied to the generator to raise the temperature of the absorbent solution. The coefficient of performance of the V.A. cycle is therefore defined by the equation:

$$C.O.P = R.E./Q_{gen} \qquad \text{---} \qquad (1.2)$$

Where

C.O.P = Coefficient of performance of the vapour absorption cycle.
R.E. = Refrigerating effect (kJ/kg).
Q_{gen} = Heat supplied to the generator (kJ/kg).

The large generator heat input required for absorption refrigeration systems usually results in a poor coefficient of performance when compared directly to compression systems. However it must be remembered that the electrical energy usually used to power compression systems is initially generated from heat. In the case of absorption systems this heat can be used directly to heat the generator whereas in compression systems the heat must first be converted to mechanical power (usually via electrical power) with large unavoidable losses. Thus if the performance of a V.A. system is to be compared fairly with that of a V.C. system all generation and transmission losses from the initial heat source to the point of use should be included.

The V.A. cycle is particularly suitable for use in industries where waste heat is available for heating the generator and or where the size of the system required would result in undesirably large compressors if a V.C. cycle were to be used.

The two main absorbent - refrigerant combinations which have been commonly used with the absorption refrigeration cycle are:

i) Lithium bromide and water with water as the refrigerant and Lithium bromide as the absorbent.

ii) Ammonia and water with ammonia as the refrigerant and water as the absorbent.

Although these absorbent refrigerant combinations are the ones most often used in actual applications many other possible combinations exist. Work performed in the 1950's by research workers of the American company du Pont has identified a number of solvents which readily absorb fluoronated hydro - carbon refrigerants [8]. Some of these substances have shown potential for use in absorption refrigeration cycles. For example dimethyl ether - tetra ethylene glycol (DME - TEG) has been shown to be a satisfactory absorbent for use with Refrigerant 22 [9,10].

Further details on the absorption refrigeration cycle can be found in Appendix A as well as references 4,5 and 6.

FIG 1.3 SIMPLE ABSORPTION REFRIGERATION SYSTEM

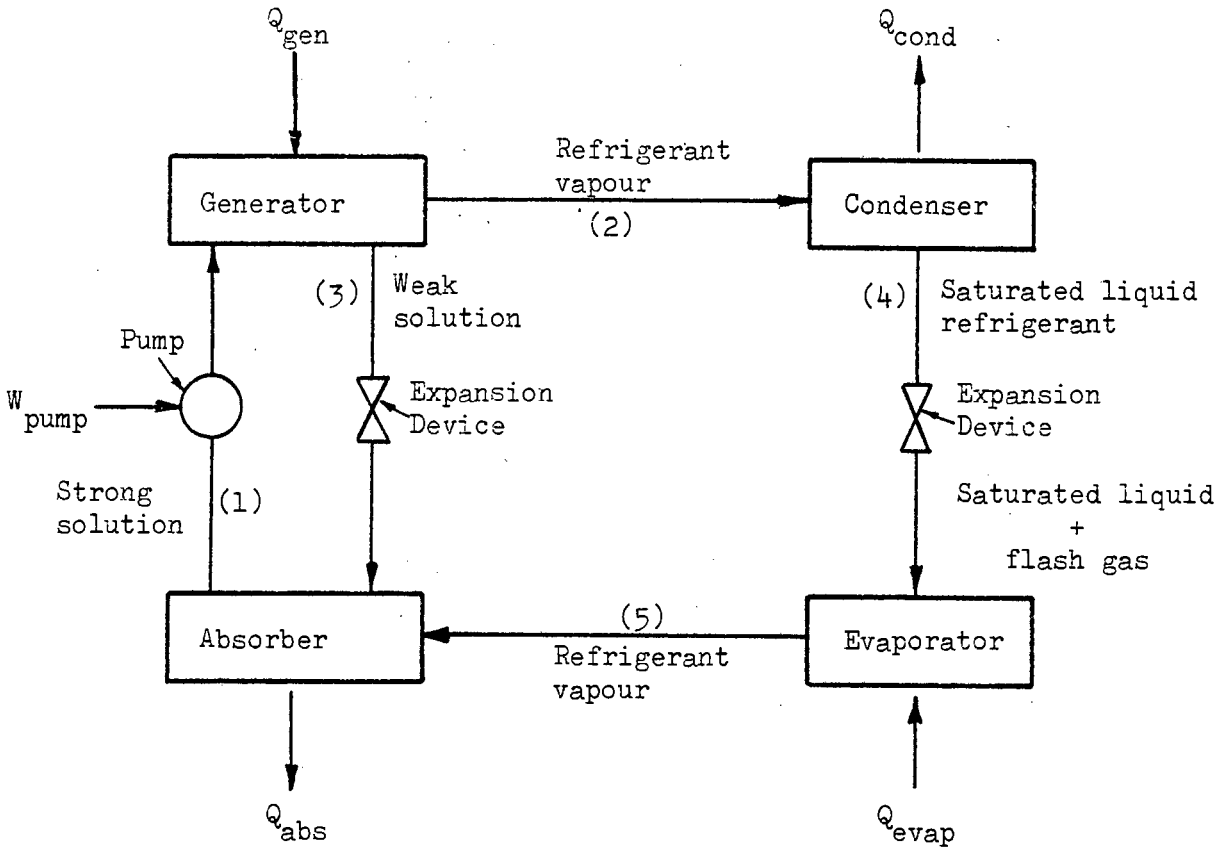
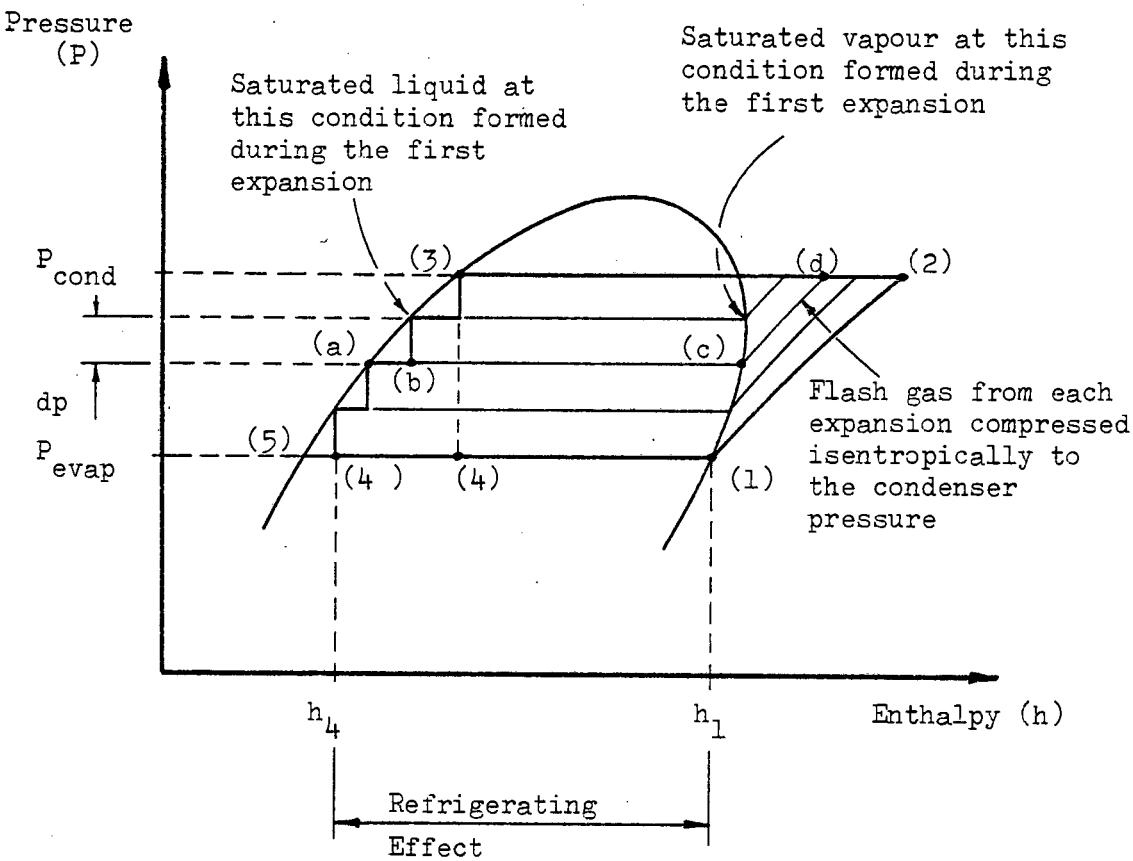


FIG 1.4 MULTIPLE EXPANSION VAPOUR COMPRESSION REFRIGERATION CYCLE ON A A PRESSURE - ENTHALPY DIAGRAM



1.5 MULTIPLE EXPANSION VAPOUR COMPRESSION CYCLES

During the expansion of the saturated liquid refrigerant from the condensing to the evaporating pressures, a portion of the refrigerant "flashes" to vapour. Since this flash gas has already undergone the liquid to vapour phase change before it enters the evaporator, it cannot produce any refrigerating effect in the evaporator. However in order to maintain the closed cycle this flash gas has to be compressed to the condensing pressure and work must be supplied to do this. This work, which is unavoidable, is performed on a portion of the refrigerant that does not produce any refrigerating effect and the result is a lower coefficient of performance than would be achieved if there was no flash gas formed during the expansion.

The expansion and compression of the single stage V.C. cycle can in principle be replaced by a series of smaller expansions and compressions as shown in Figure 1.4. Refering to this diagram, saturated liquid at the condensing pressure, i.e. condition (3), is expanded through a small pressure drop dp to a pressure of P_{cond} - dp. During this expansion some of the liquid flashes to vapour and a mixture of saturated liquid and saturated vapour results. The vapour is separated from the liquid and independantly compressed to the condenser pressure. The remaining liquid, which is now at a lower enthalpy than before the expansion, is expanded through a further pressure drop of dp. The flash gas formed during this expansion is again separated from the liquid and compressed to the condenser pressure while the liquid is expanded further. This process is repeated until the pressure of the refrigerant becomes equal to the desired evaporator pressure. The refrigerant is then evaporated in the usual way to produce the refrigerating effect.

The result of this series of expansions and compressions is that less of the refrigerant vapour has to be compressed

through the maximum system pressure difference i.e. P_{cond} - P_{evap}. Only vapour which has undergone a phase change in the evaporator would have to be compressed from the lowest to the highest system pressure. Less compressor work would therefore be required for a given refrigerating effect. This reduction in the compressor work would result in an improved coefficient of performance.

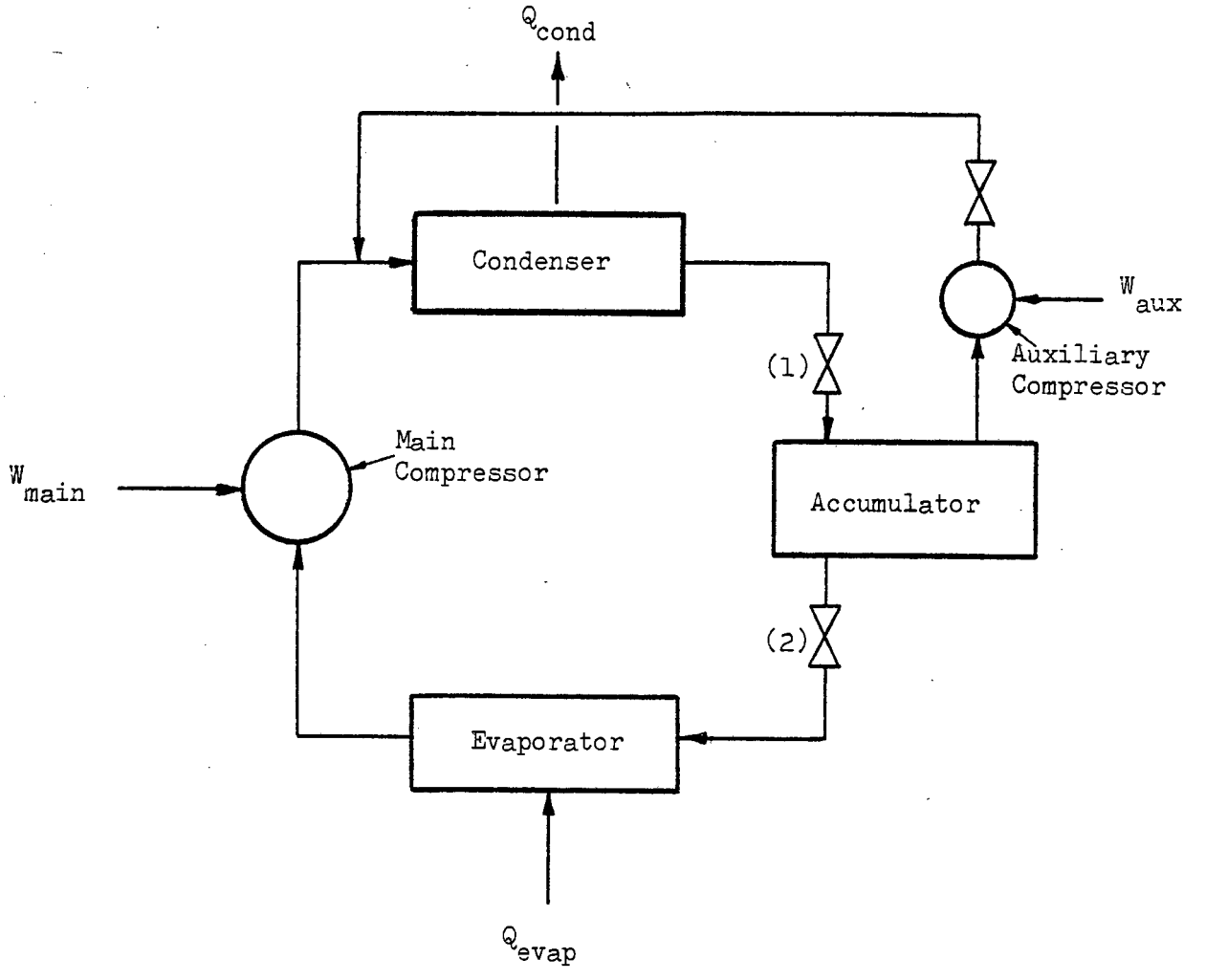
Furthermore since the enthalpy of the refrigerant entering the evaporator is reduced from what it would be with a single expansion (see Figure 1.4), the refrigerating effect is increased i.e. For the same refrigerant flow rate in the evaporator more heat would be absorbed.

A refrigeration cycle has been described by Van Breda Smith [7] which achieves the progressive expansion of the refrigerant, as described above, on a batch process basis. A sketch of this system is shown in Figure 1.5. The major components of the system are an evaporator, a condenser, an accumulator, a main compressor and an auxiliary compressor.

In this system, as in conventional V.C. systems, refrigerant is continuously evaporated in the evaporator to produce the refrigerating effect. This vapour is compressed by the main compressor and passed to the condenser where it is condensed.

The system differs from the conventional V.C. system in the method of expanding the saturated liquid refrigerant from the condensing to the evaporating pressure. Instead of the usual constant enthalpy expansion device, an auxiliary expansion system is employed. This auxiliary system collects a batch of the saturated liquid refrigerant from the condenser and progressively reduces its pressure, starting at the condenser pressure and ending at the evaporator pressure. At the start of a batch, saturated liquid from the condenser sump is transfered to the accumulator at constant

FIG 1.5 DIFFERENTIAL BATCH EXPANSION AND COMPRESSION REFRIGERATION SYSTEM



pressure. For this operation valve (1) is opened and valve (2) is closed. Once the accumulator has been fully charged, valve (1) is closed and the auxiliary compressor begins to reduce the pressure of the refrigerant in the accumulator. Flash gas is formed as each induction stroke of the compressor causes a lowering of the pressure in the accumulator. This flash gas is compressed by the auxiliary compressor from the pressure at which it is formed to the condenser pressure. Once the pressure in the accumulator reaches the evaporator pressure, valve (2) is opened and the refrigerant flows at constant pressure into the evaporator where it is evaporated to produce the refrigerating effect. When the accumulator has discharged all of the refrigerant, valve (2) is closed and the expansion process is repeated on another batch of refrigerant from the condenser.

The differential batch expansion system, as this system has been called, is an effective batch process multiple expansion and compression refrigeration cycle. Each stroke of the auxiliary compressor causes a small reduction in the pressure in the accumulator and this results in a small expansion of the refrigerant. The size and speed of the auxiliary compressor determines the number of expansions achieved and the compressor friction losses suffered. A small volume high speed auxiliary compressor would result in a large number of expansions but high compressor friction losses while a large volume low speed auxiliary compressor would result in a smaller number of expansions but lower friction losses. Clearly the size and speed of the auxillary compressor should be chosen so that the extra friction losses in the auxiliary compressor do not outweigh any gain in performance resulting from the multiple expansions.

Although the system effectively achieves a very large number of expansions it is however a batch process system which is generally less desirable than a continuous process system for industrial applications. From a practical point of view

the differential batch expansion system might be improved for an actual installation by the introduction of a second accumulator operating in parallel to the existing one. This would allow one accumulator to be charged or discharged while the other was being decompressed. This would enable the auxiliary compressor to operate continuously, merely switching to the other accumulator as it finished decompressing the first.

The multiple expansion vapour compression cycle, as represented in Figure 1.4, would require special arrangements to effect the compression of the flash gas from each expansion stage. Two possible options which could be considered to compress the flash gas to the condenser pressure are:

i) To use several small auxiliary compressors where each compressor is dedicated to one stage and designed to operate between the stage pressure and the condenser pressure. Such a system would suffer from a high initial cost due to the large number of compressors required. In addition, since the friction losses of each compressor would be similar to those of the much larger main compressor, it is probable that any gain in performance due to multiple expansions would be lost to increased compressor friction.

ii) To use a single auxiliary compressor which had a single discharge at the condenser pressure and which was capable of compressing vapour from many different suction pressures corresponding to the pressures at each expansion stage. Such a compressor would require considerable development and thus this option cannot be readily achieved in practice.

In view of the difficulties of compressing the flash gas formed in a multiple expansion and compression cycle, a

refrigeration cycle was proposed in which a small auxiliary absorption system was combined with a multiple expansion V.C. cycle. The absorption part of the cycle would absorb the flash gas from each expansion and return it to the condenser using standard absorption techniques. The compressor would then be left to compress the bulk of the refrigerant vapour, i.e. the vapour evaporated in the evaporator to produce a refrigerating effect, to the condenser pressure. Figure 1.6 shows a simplified flow diagram of the proposed mulptiple expansion - absorption refrigeration cycle. With this cycle it was expected that the compressor work performed per unit of refrigeration would be reduced since it would no longer be compressing the flash gas as in a normal V.C. cycle. The absorption side of the cycle would require little mechanical work with the main energy input to this part of the cycle being heat. It was therefore anticipated that, if this heat was available as waste energy which would otherwise be rejected to the environment and could thus be considered as "free", savings in high grade energy would be realised with the combined compression - absorption system. In order to be able to use a fluoronated hydro - carbon refrigerant it was proposed to use dimethyl ether - tetra ethylene glycol (DME - TEG) as the absorbent. This solvent, as previously mentioned, has been shown [9,10] to be a satisfactory absorbent for use with difluoro chluoro methane (R22) as refrigerant.

CHAPTER 2

THEORETICAL ASPECTS OF MULTIPLE EXPANSION CYCLES

2.1 PERFORMANCE LOSS DUE TO FLASH GAS FORMATION

The formation of flash gas during the expansion of the refrigerant in any refrigeration cycle leads to a lower coefficient of performance for the cycle than would be achieved if no flashing to vapour occured. This is because the flash gas, which cannot produce any refrigerating effect, still has to have work done on it to return it to the condenser. This work is expended on a non productive portion of the refrigerant and this reduces the performance.

Figure 2.1 shows a simple refrigeration cycle with a constant enthalpy expansion on a P-h diagram. From energy and mass balances on the refrigerant it can be shown that the fraction of the refrigerant that flashes to vapour in the expansion is given by the equation :

$$x = (h_2-h_3)/(h_4-h_3) \qquad \text{---} \qquad (2.1)$$

Where

x = Vapour fraction.
h_i = Refrigerant enthalpy at cycle point i (kJ/kg).
i = Cycle point number as shown on Figure 2.1.

The flash gas mass flow rate and the compressor work required to recompress the vapour to the original pressure are given by the equations :

$$m_f = x \cdot m_1 \qquad \text{---} \qquad (2.2)$$

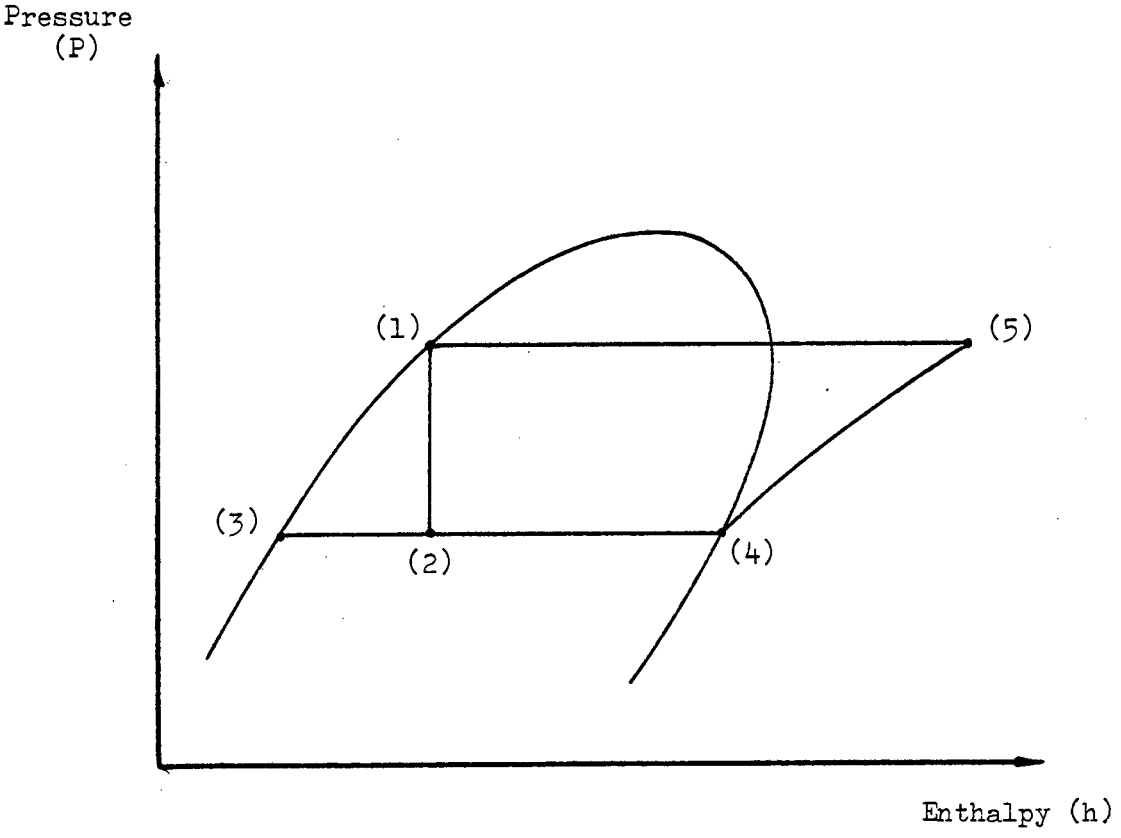
$$W_f = m_f \cdot (h_5 - h_4) \qquad \text{---} \qquad (2.3)$$

Where

m_f = Flash gas mass flow rate (kg/s).
m_1 = Refrigerant mass flow rate at point 1 (kg/s).
W_f = Compressor work required to compress the flash gas (kJ/s).

The flash gas takes up compressor capacity (in the case of a V.C. cycle) and absorber/generator capacity (in the case of an absorption cycle) which could otherwise, if there was no flash gas, be used for refrigerant vapour which had produced a refrigerating effect. The refrigerating capacity of the sytem is thus reduced by the existence of the flash gas.

FIG 2.1 SIMPLE VAPOUR COMPRESSION REFRIGERATION CYCLE ON A PRESSURE - ENTHALPY DIAGRAM



2.2 THE MULTIPLE EXPANSION VAPOUR COMPRESSION CYCLE

The multiple expansion and compression cycle can be analysed theoretically using a similar approach to that used for the single stage V.C. cycle. Figure 2.2 shows the flow diagram of the cycle with various points in the cycle numbered for easy reference. The numbered points correspond with those in Figure 2.3 which shows the same cycle represented on a P-h diagram. The equations which describe the cycle are presented below. Each item in the system has been considered individually and the subscripts refer to the numbered points shown in Figures 2.2 and 2.3.

For the first expansion :

$$x_1 = (h_2-h_3)/(h_4-h_3) \quad --- \quad (2.4)$$

$$m_4 = x_1 \cdot m_1 \quad --- \quad (2.5)$$

$$m_3 = (1-x_1) \cdot m_1 \quad --- \quad (2.6)$$

$$W_1 = m_4 \cdot (h_{18}-h_4) \quad --- \quad (2.7)$$

Where

x_1 = Vapour fraction for first stage.
h_i = Refrigerant enthalpy at cycle point i (kJ/kg).
m_i = Refrigerant mass flow rate at cycle point i (kg/s).
W_1 = Compressor work to compress the flash gas from the first stage (kJ/kg).

Similar equations to these can be developed for each of the expansion stages and enable the mass flow rates and compressor work for every stage to be determnined.

FIG 2.2 MULTIPLE EXPANSION VAPOUR COMPRESSION REFRIGERATION CYCLE FLOW DIAGRAM

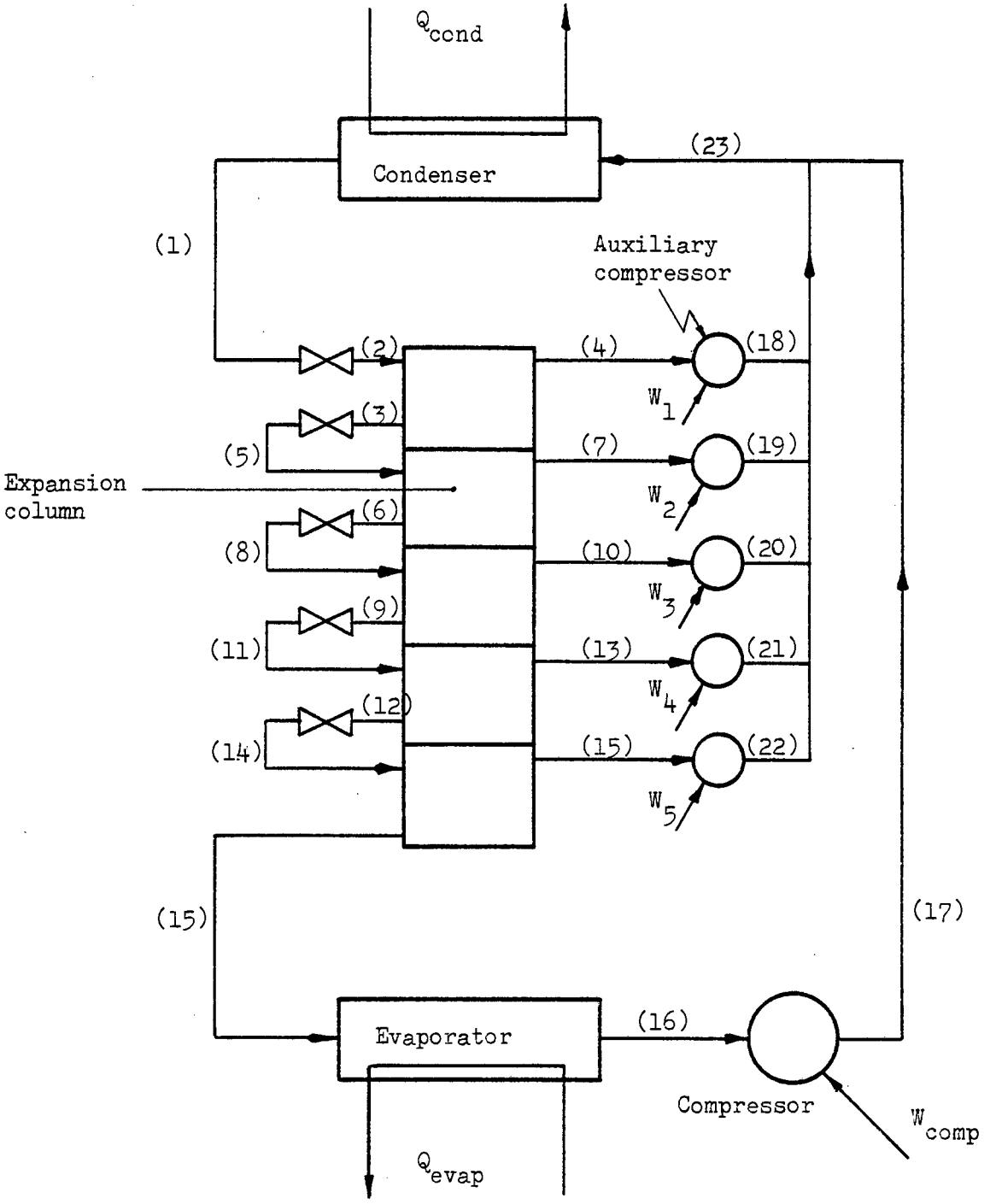
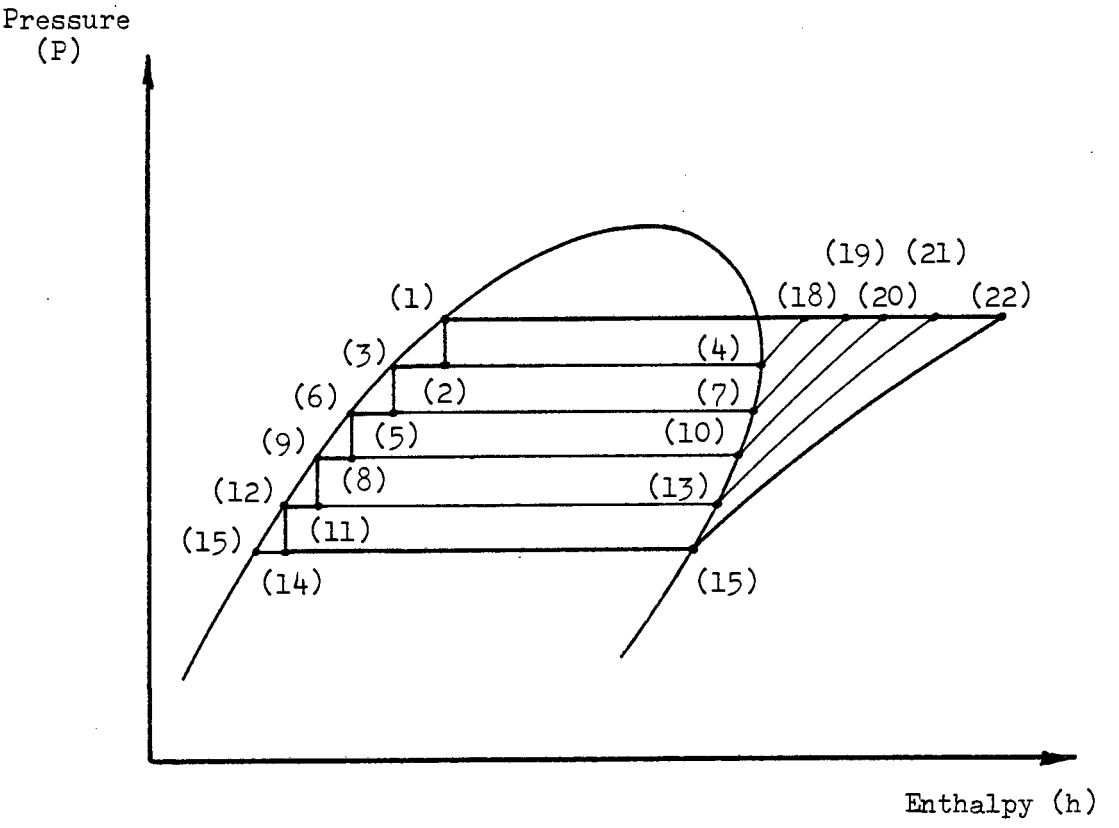


FIG 2.3 MULTIPLE EXPANSION VAPOUR COMPRESSION REFRIGERATION CYCLE ON A PRESSURE - ENTHALPY DIAGRAM



For the evaporator :

$$Q_{evap} = m_{15} \cdot (h_{16} - h_{15}) = m_{we} \cdot Cp_w \cdot dT_{we} \quad --- \quad (2.8)$$

Where

Q_{evap} = Heat absorbed in the evaporator (kJ/s).
Cp_w = Specific heat capacity of water (kJ/kg K).
m_{we} = Mass flow rate of evaporator heating water (kg/s).
dT_{we} = Evaporator heating water temperature change (K)

For the compressor :

$$W_{comp} = m_{16} \cdot (h_{17} - h_{16}) \quad --- \quad (2.9)$$

Where

W_{comp} = Main compressor work (kJ/s).

For the system :

The coefficient of performance of the system can be defined by the equation :

$$C.O.P. = Q_{evap} / W_{total}$$

$$= Q_{evap} / \left[W_{comp} + \sum_{j=1}^{5} W_j \right] \quad --- \quad (2.10)$$

Where

C.O.P. = Coefficient of performance of the multiple expansion and compression cycle.

W_j = Auxiliary compressor work for the j^{th} stage (kJ/s).

From Figure 2.3 it can be seen that to compress flash gas from the first stage to the condensing pressure would require an ideal work input of $(h_{18}-h_4)$ kJ/kg. To compress vapour from the last stage to the condensing pressure would require a work input of $(h_{22}-h_{15})$ kJ/kg. Less compressor work is therefore required to compress flash gas from the stage pressures than from the lower evaporating pressure to the condensing pressure. It can therefore be predicted that the theoretical performance of the multiple expansion and compression cycle would be better than that of a single stage cycle.

Full details of a theoretical analysis of the cycle which was performed using the above equations encoded into a computer programme have been presented in appendix B.

2.3 THE MULTIPLE EXPANSION - ABSORPTION CYCLE

The multiple expansion - absorption cycle can also be theoretically analysed using methods similar to those used for other refrigeration cycles. Figure 2.4 shows the cycle with various points numbered for easy reference. The equations for the system have been developed below using subscripts which refer to the numbered cycle points in Figure 2.4. For simplicity each item in the system has been considered individually.

For the expansion column :

The expansion process performed in the expansion column has been shown on a p-h diagram for the refrigerant in Figure 2.5. In each stage saturated liquid is expanded with constant enthalpy to a lower pressure. In the process flash gas and saturated liquid (at a lower enthalpy) is formed.

Considering the first expansion the following equations can be developed :

$$x_1 = (h_2-h_3)/(h_4-h_3) \quad --- \quad (2.11)$$

$$m_4 = x_1 \cdot m_1 \quad --- \quad (2.12)$$

$$m_3 = (1-x_1) \cdot m_1 \quad --- \quad (2.13)$$

Where

x_1 = Vapour fraction at stage 1.
h_i = Refrigerant enthalpy at cycle point i (kJ/kg).
m_i = Mass flow rate at cycle point i (kg/s).
i = Cycle point as shown on Figures 2.4 and 2.5.

Similar equations can be developed for all the stages of the expansion.

FIG 2.4 MULTIPLE EXPANSION - ABSORPTION REFRIGERATION CYCLE FLOW DIAGRAM

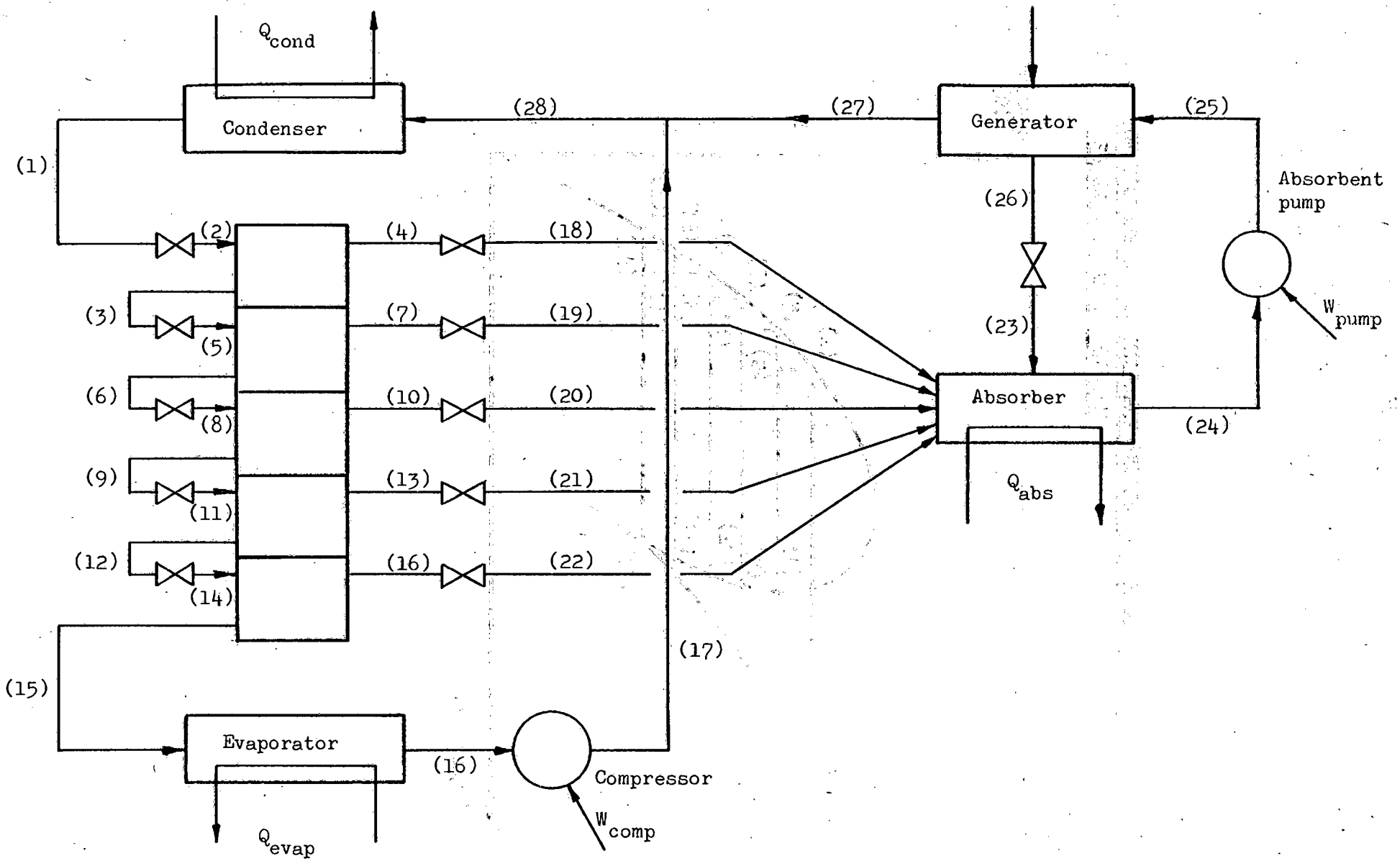
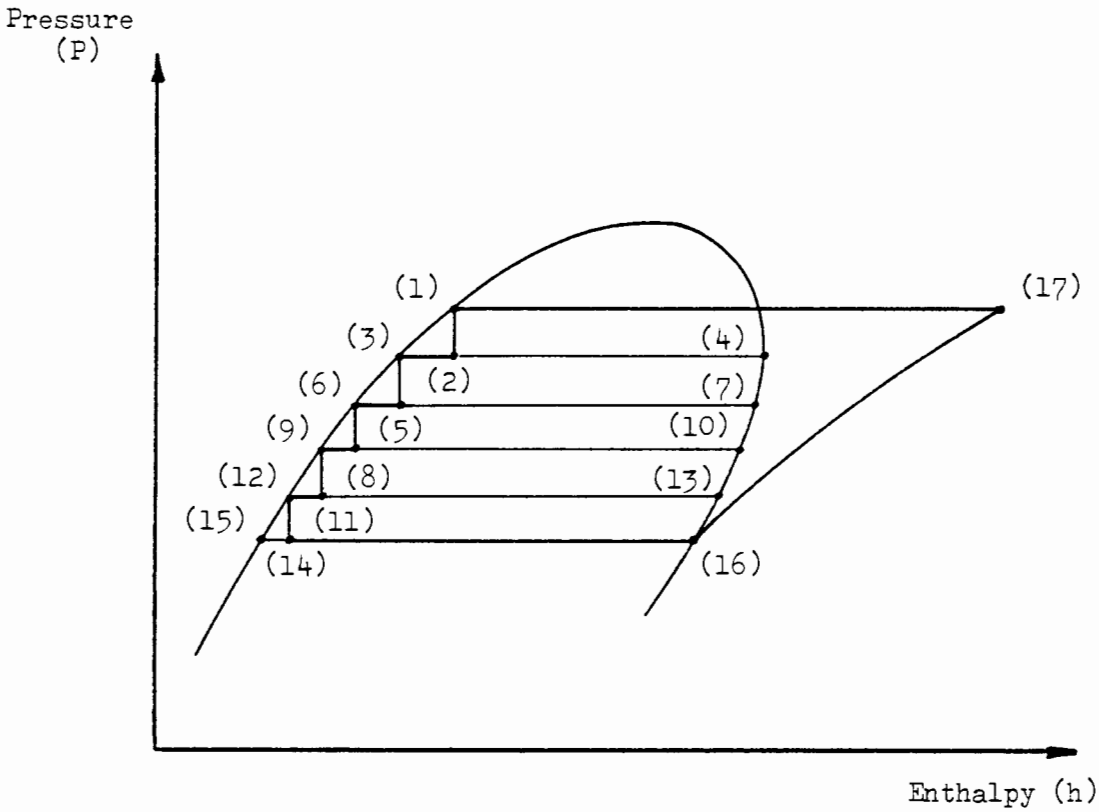


FIG 2.5 EXPANSION PROCESS OF THE MULTIPLE EXPANSION - ABSORPTION CYCLE ON A PRESSURE - ENTHALPY DIAGRAM



For the last expansion :

$$x_5 = (h_{14}-h_{15})/(h_{16}-h_{15}) \quad --- \quad (2.14)$$

$$m_{15} = (1-x_5)\cdot m_{12} \quad --- \quad (2.15)$$

$$m_{16} = x_5\cdot m_{12} \quad --- \quad (2.16)$$

Substituting for m_{12} etc in (2.15) and (2.16) gives :

$$m_{15} = (1-x_1)\cdot(1-x_2)\cdot(1-x_3)\cdot(1-x_4)\cdot(1-x_5)\cdot m_1 \quad -- \quad (2.17)$$

$$m_{16} = x_1\cdot x_2\cdot x_3\cdot x_4\cdot x_5\cdot m_1 \quad -- \quad (2.18)$$

For the evaporator :

The heat absorbed in the evaporator is given by the usual equation :

$$Q_{evap} = m_{15}\cdot(h_{16}-h_{15}) = m_{we}\cdot Cp_w\cdot dT_{we} \quad --- \quad (2.19)$$

Where

Q_{evap} = Heat absorbed in the evaporator (kJ/s).
m_{we} = Heating water mass flow rate (kg/s).
Cp_w = Specific heat capacity of water (kJ/kg K).
dT_{we} = Evaporator heating water temperature change (K).

For the compressor :

The theoretical work input to the compressor is given by the equation :

$$W_{comp} = m_{16}\cdot(h_{17}-h_{16}) \quad --- \quad (2.20)$$

Where

W_{comp} = Compressor work (kJ/s).

For the absorber :

The absorber can be analysed by considering mass and energy balances on the vessel. Although the absorber would not in practice be insulated it has been assumed for simplicity that all heat transfer is to the cooling water.

Overall mass balance gives :

$$m_{24} = m_{18} + m_{19} + m_{20} + m_{21} + m_{22} + m_{23} \quad -- \quad (2.21)$$

Energy balance gives :

$$m_{wa} \cdot Cp_w \cdot dT_{wa} = \sum_{j=18}^{23} m_j \cdot h_j + m_{24} \cdot h_{24} \quad -- \quad (2.22)$$

Refrigerant mass balance gives :

$$m_{24} \cdot C_{24} = m_{23} \cdot C_{23} + \sum_{j=18}^{22} m_j \quad --- \quad (2.23)$$

Where

m_{wa} = Absorber cooling water flow rate (kg/s).
dT_{wa} = Absorber cooling water temperature change (K).
Cp_w = Specific heat capacity of water (kJ/kg K).
C_i = Refrigerant concentration in the absorbent at cycle point i (% by mass).

For the generator :

The generator can similarly be analysed by considering mass and energy balances.

Overall mass balance gives :

$$m_{25} = m_{27} + m_{26} \qquad \text{---} \quad (2.24)$$

Energy balance (assuming perfect insulation)gives :

$$Q_{gen} = m_{26} \cdot h_{26} + m_{27} \cdot h_{27} + m_{25} \cdot h_{25} \qquad \text{---} \quad (2.25)$$

Refrigerant mass balance gives :

$$m_{27} = m_{25} \cdot C_{25} - m_{26} \cdot C_{26} \qquad \text{---} \quad (2.26)$$

Where

Q_{gen} = Heat supplied to the generator (kJ/s).
C_i = Concentration of refrigerant in the absorbent at cycle point i (kg/kg).

For the pump :

The work input to the pump is given by the equation :

$$W_{pump} = m_{24} \cdot (h_{25} - h_{24}) \qquad \text{---} \quad (2.27)$$

Where

W_{pump} = Work input to absorbent pump (kJ/s).

For the condenser :

Assuming the condenser to be perfectly insulated the following equations can be developed :

$$Q_{wc} = m_{wc} \cdot Cp_w \cdot dT_{wc} = m_{28} \cdot (h_{28} - h_1) \quad --- \quad (2.28)$$

Assuming that no heat losses occur in the pipework connecting the condenser, generator and compressor an energy balance of the mixing of the two refrigerant flows gives :

$$m_{17} \cdot h_{17} = m_{27} \cdot h_{27} + m_{28} \cdot h_{28} \quad --- \quad (2.29)$$

Also :

$$m_{28} = m_{27} + m_{17} \quad --- \quad (2.30)$$

For the system :

Depending on the particular application of the cycle, the circumstances relating to the supply of "waste" energy and what is considered to be the energy "input" to the system the coefficient of performance could be defined in a number of ways. For the purposes of this theoretical analysis all energy inputs to the cycle are considered and the coefficient of performance is then defined as :

$$C.O.P. = Q_{evap} / [W_{comp} + W_{pump} + Q_{gen}] \quad -- \quad (2.31)$$

Where

C.O.P. = Coefficient of performance of multiple expansion - absorption cycle.

The enthalpies required for these calculations are obtained from the Pressure - enthalpy diagram for the refrigerant and from the concentration - enthalpy diagram for the refrigerant absorbent combination. The mass flow rates required are obtained from the design specifications of the required refrigerating capacity and the relevant energy and mass balance equations.

Since concentration - enthalpy diagrams for R22 and dimethyl ether - tetra ethylene glycol could not be obtained it was not possible to perform these calculations and thereby determine a theoretical COP for the cycle. An experimental approach was therefore followed to determine the performance of the cycle.

CHAPTER 3

EXPERIMENTAL APPARATUS

3.1 GENERAL DESCRIPTION

The multiple expansion - absorption experimental apparatus consisted of an evaporator, condenser, expansion column, compressor, absorber, generator, heat exchanger and pump unit arranged as shown in the system flow diagram of Figure 3.1.

Saturated liquid refrigerant from the condenser was passed to the expansion column where it was expanded in five stages to the evaporating pressure. The low presssure saturated liquid from the last stage of the expansion column passed into the bottom of the evaporator where it was evaporated to produce the refrigerating effect. Refrigerant vapour was extracted from the top of the evaporator and compressed to the condensing pressure by the compressor.

Flash gas formed with each expansion was separated from the liquid in the expansion chamber and passed via a pressure reducing valve to the absorber where it was absorbed. Refrigerant laden absorbent (strong solution) was pumped from the absorber to the generator where it was heated by an electric element. The refrigerant vapour released from the strong solution in the generator was passed to the condenser where it was condensed along with the refrigerant vapour from the compressor. A non return valve allowed vapour to "blow off" from the generator as the pressure there became higher than the pressure in the condenser while preventing

the flow of vapour from the compressor into the generator. Refrigerant depleted absorbent (weak solution) flowed from the generator back to the absorber through a regenerative heat exchanger where it partially cooled in the process of pre - heating the cold absorbent being pumped to the generator . Cooling coils in the top section of the absorber cooled to nearly ambient temperature the warm absorbent returning to the absorber so enabling it to re-absorb refrigerant vapour. A throttling valve was used to maintain the pressure difference between the generator and the absorber.

Plate 3.1 shows a photograph of the experimental apparatus which was constructed on a tubular steel framework with chipboard back and centre panels. The apparatus was arranged with the absorption equipment i.e. generator, absorber and pump unit all located on one side of a centre dividing panel. The remaining items were located on the other side and the rig was therefore roughly divided into an absorption and a compression side. The various vessels in the system were mounted onto the back and centre panels with appropriate brackets while the compressor and pump unit were mounted onto steel base plates across the bottom of the frame. Detailed descriptions and constructional details of each piece of equipment have been given under separate headings below.

FIG 3.1 MULTIPLE EXPANSION - ABSORPTION SYSTEM FLOW DIAGRAM

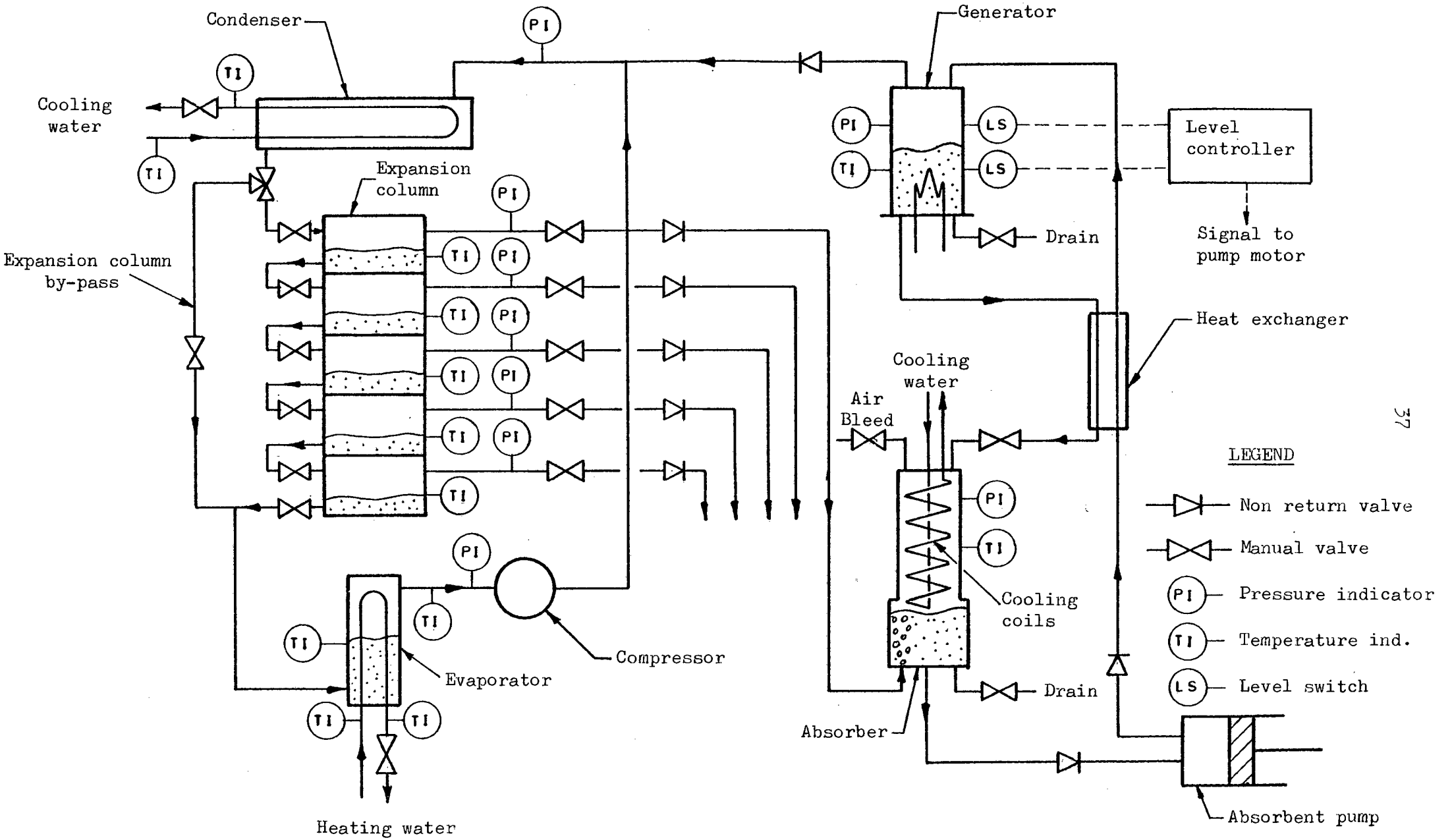
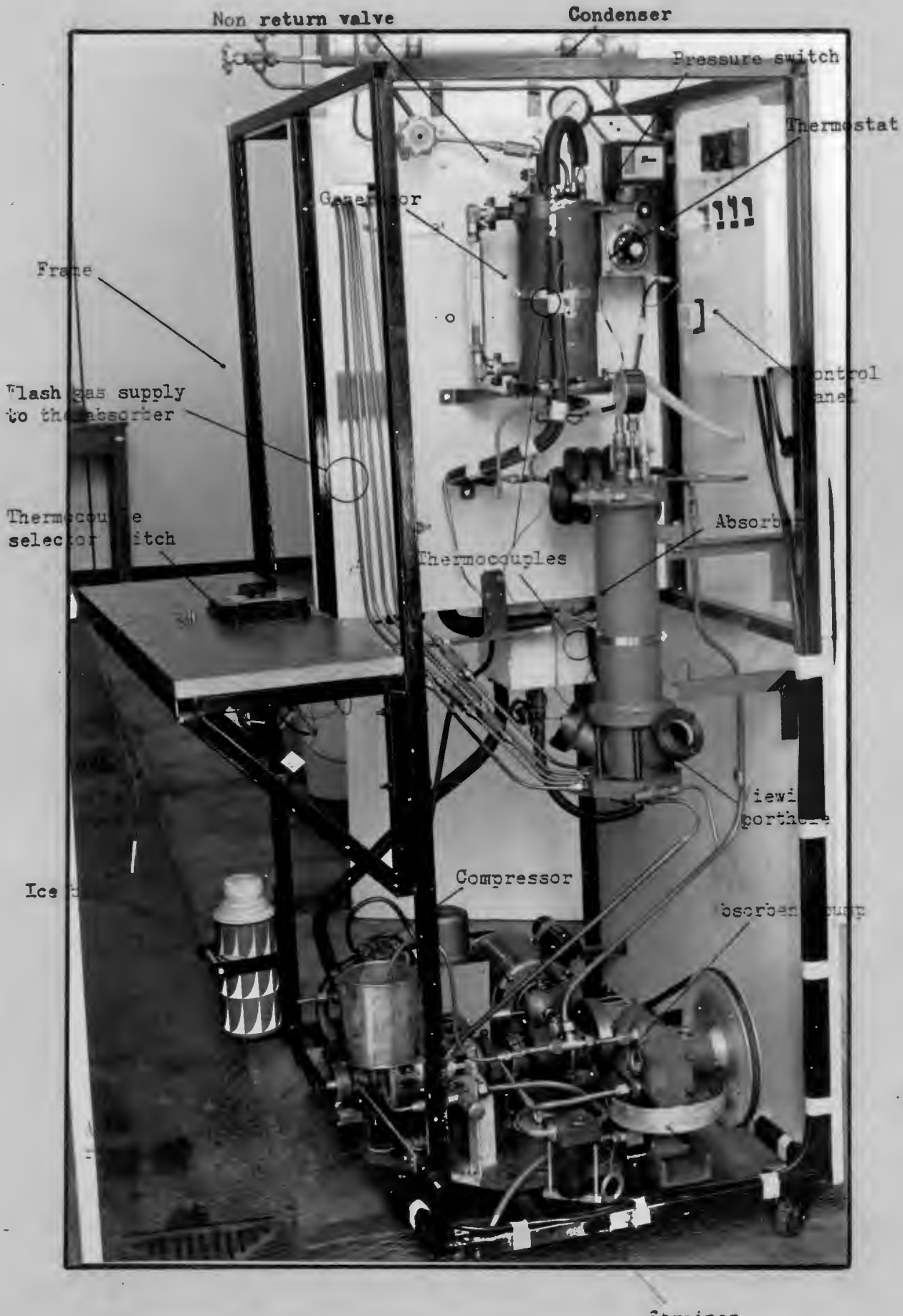


PLATE 3.1 GENERAL VIEW OF THE EXPERIMENTAL APPARATUS



3.2 THE EXPANSION COLUMN

Figure 3.2 shows the flow diagram for the expansion column. Saturated (or sometimes subcooled) liquid refrigerant from the condenser was expanded through a throttling valve into the top most chamber of the column. In this chamber the flash gas separated from the liquid enabling it to be bled off from the top of the chamber. This flash gas was then passed via a throttling valve to the absorber. The remaining saturated liquid, which collected at the bottom of the chamber, was expanded through the next expansion valve to the chamber below where the process of vapour separation and removal was repeated. The refrigerant liquid continued passing down the column until it reached the lowest chamber where the pressure was equal to the evaporator pressure. The refrigerant then flowed from this chamber to the evaporator at constant pressure. The pressure in each chamber was measured with a pressure gauge connected into the chamber while the temperature was measured with a thermocouple attached to the outside of the chamber wall with a jubilee clamp.

The expansion column was made up of five similar chambers which were designed to stack one on top of the other so that the bottom of each chamber formed the cover for the chamber below. Each chamber was constructed of a steel pipe closed at the lower end by a welded blank flange. The open upper end had a welded rim which was complete with an "O" ring groove for sealing between the chambers. The five chambers were held together by means of clamping plates at the top and bottom which were pulled together by means of threaded tie rods. This modular construction enabled the column to be dismantled easily for the various modifications and changes which were effected.

Each chamber was manufactured complete with the expansion valve, flash gas throttling valve, pressure gauge connection and fluid passages in the bottom flange which was made thick enough for this purpose. All the fittings and valves were silver soldered directly into the flanges to minimize the number of joints where leaks could occur.

Figure 3.3 shows a general assembly of the top most expansion chamber and its cover plate which was only required for the top chamber. All of the other chambers were similar with the exception of the bottom chamber which did not have those features in the bottom flange which were required for the chamber below.

Plate 3.2 shows a photograph of the assembled five stage expansion column with the insulation removed. Clearly visible in the photograph are the expansion valves, flash gas throttling valves, clamping plates, tie rods and pressure gauges.

FIG 3.2 EXPANSION COLUMN FLOW DIAGRAM

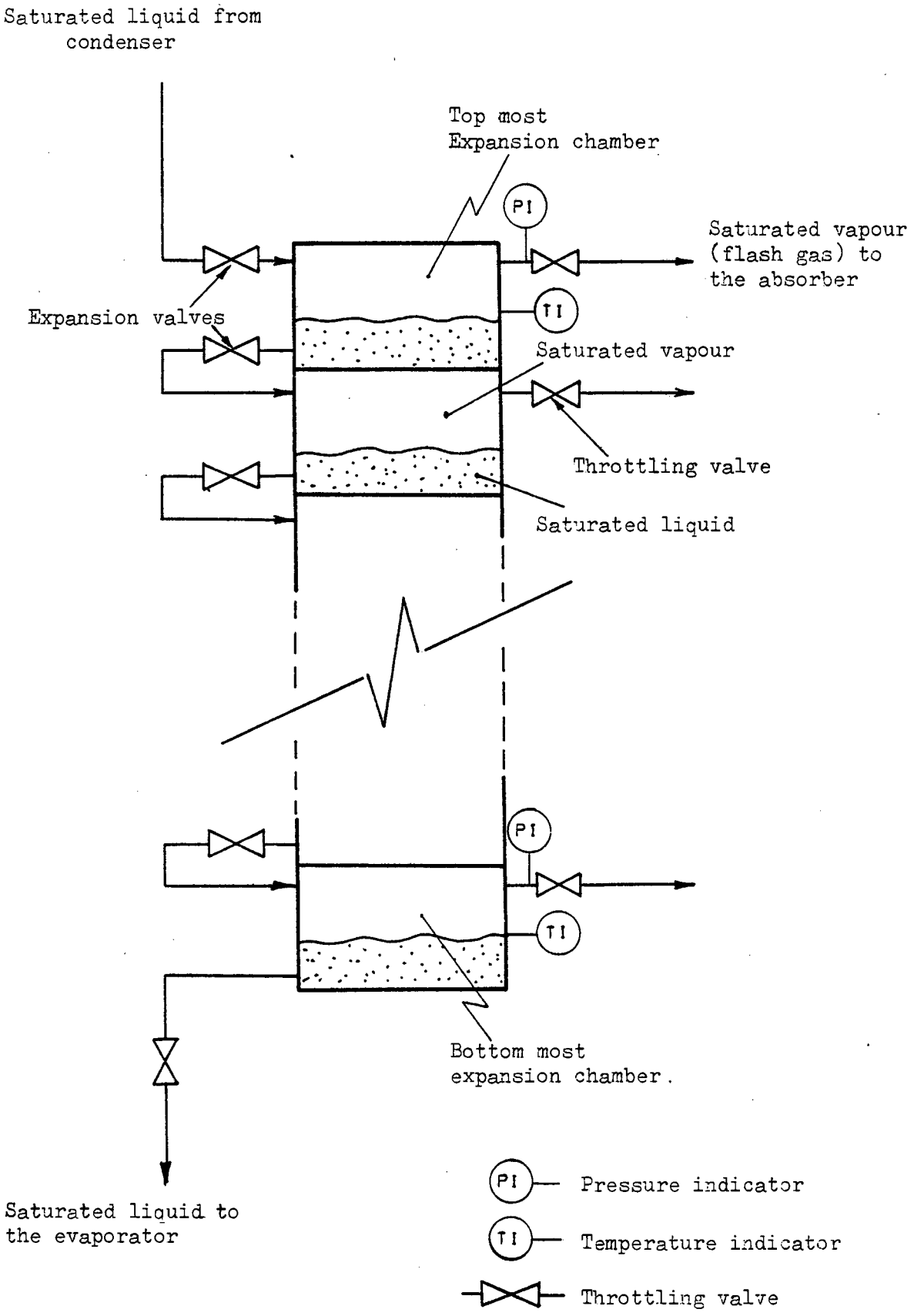
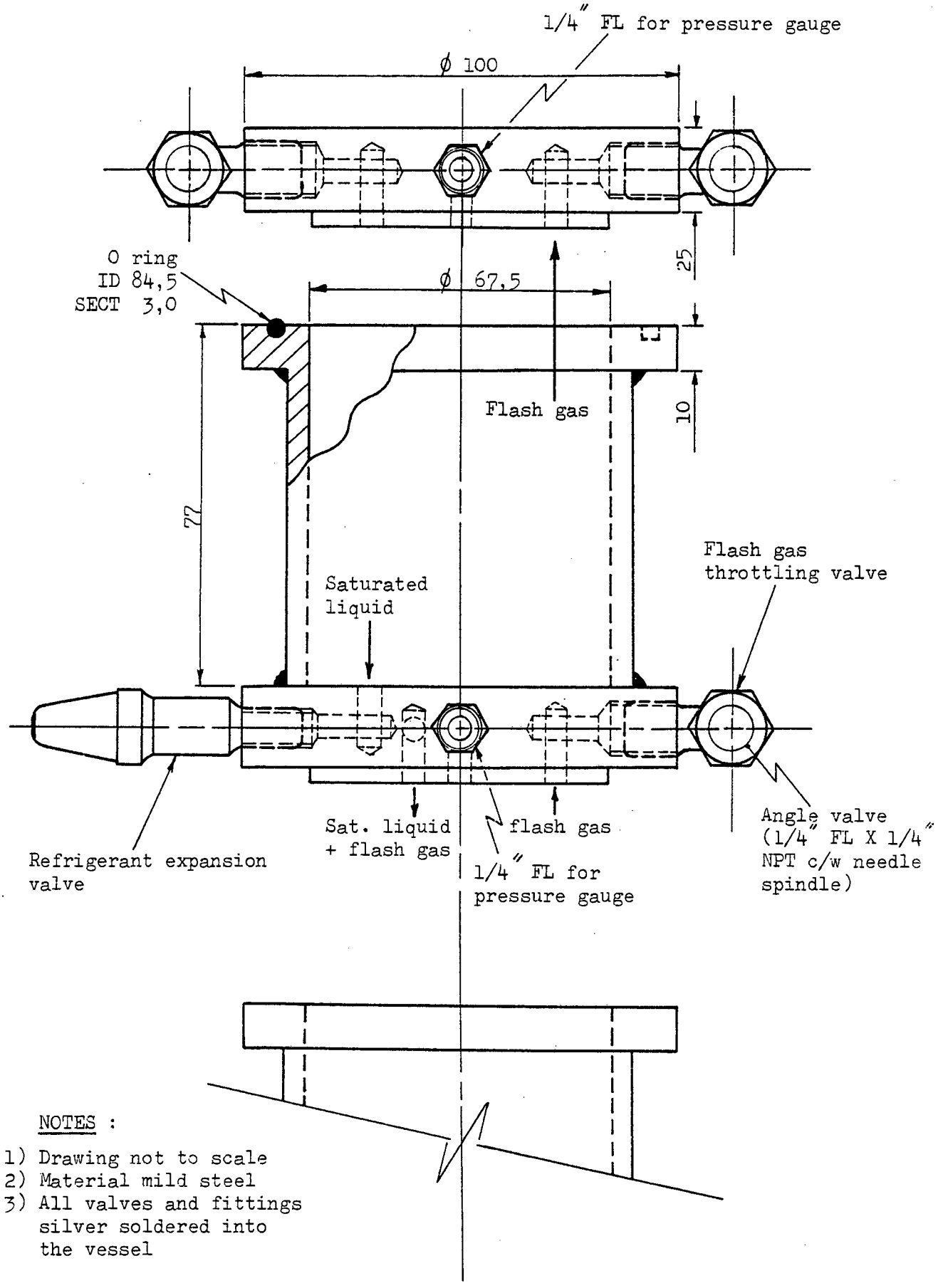


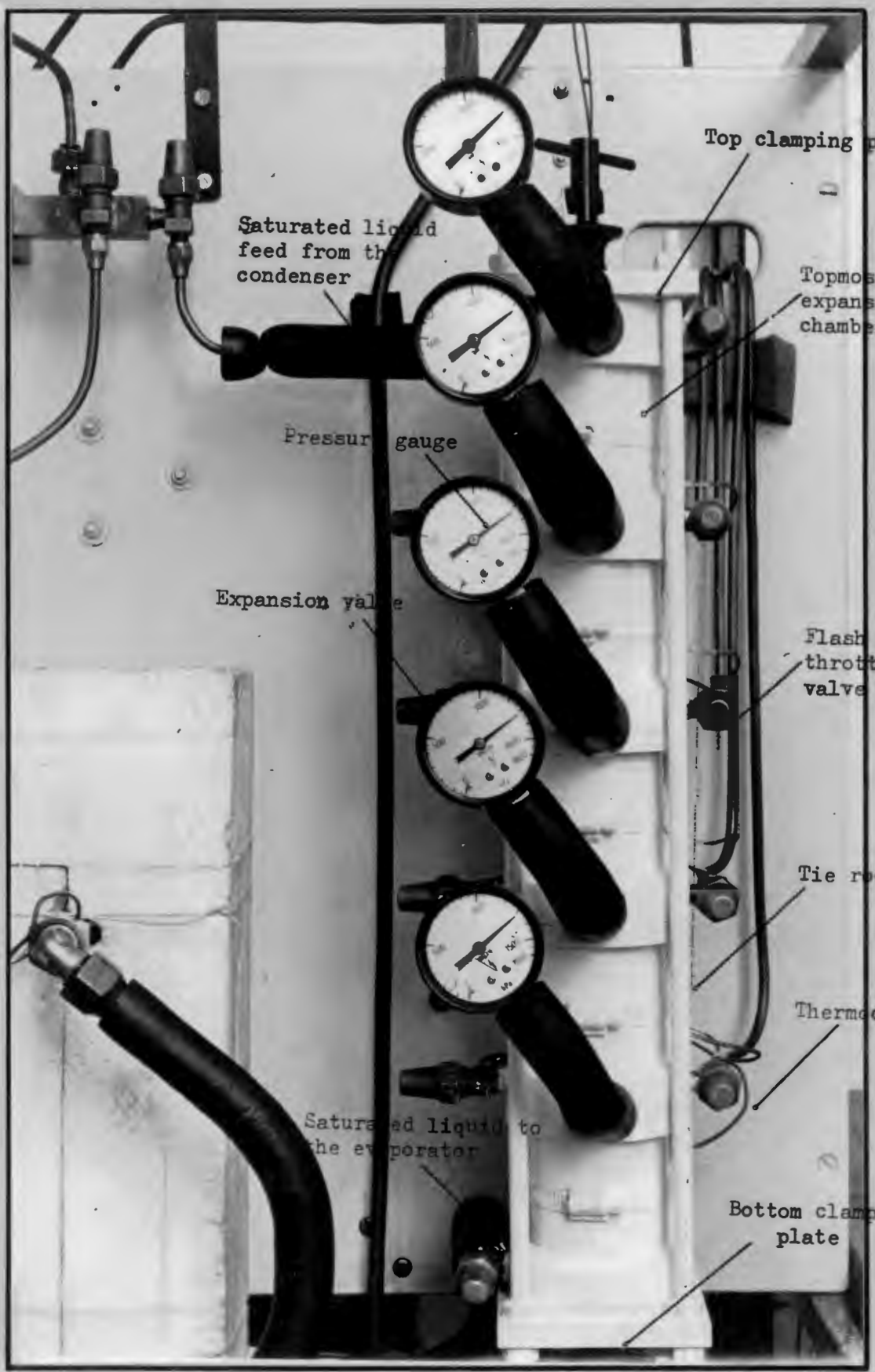
FIG 3.3 GENERAL ASSEMBLY OF EXPANSION CHAMBER AND TOP PLATE



NOTES :

1) Drawing not to scale
2) Material mild steel
3) All valves and fittings silver soldered into the vessel

PLATE 3.2 ASSEMBLED EXPANSION COLUMN WITH THE INSULATION REMOVED



3.3 THE GENERATOR

Figure 3.4 shows the flow diagram of the generator which was in essence a cylindrical steel vessel equiped with an electric heating element in the bottom. Strong absorbent solution in the vessel was heated by the element and the refrigerant driven off. The released vapour collected in the top of the vessel from where it passed out of the generator to the condenser. The weak solution remaining, being of higher density than the strong solution, migrated due to gravity to the bottom of the generator from where it flowed out of the generator to the absorber. The temperature in the generator was controlled by a thermostat which operated the heating element. The liquid level was maintained between set levels by a simple level controller which started and stopped the absorbent pump when high and low level switches in the vessel were activated. A pressure switch which switched off the pump and the heating element at the maximum safe generator pressure was fitted to the vessel to provide over pressure protection.

Figure 3.5 shows a cross sectioned sketch of the generator vessel which was constructed from a steel pipe (int. dia. 100mm and length 250mm) which was closed at the top and flanged at the bottom. A spiral heating element (220 V, 1 kW), a drain valve and a liquid outlet connection were fitted into the blank flange which formed the bottom of the vessel. A sight glass which indicated the liquid level in the vessel was fitted to the side of the vessel and a level control unit was inserted into the vessel through the top.

The level control unit consisted of two copper tubes which were sealed at the lower end and at the fixing flange and which projected down into the vessel. A copper float with an open top was constrained to slide up and down the outside of the tubes. Reed switches which were embedded in "Tufnol" rods inside the copper tubes were arranged so that they

could be activated by a small magnet affixed to the float. Provision was made to raise and lower the rods containing the reed switches which enabled the set high and low liquid levels to be adjusted easily. The reed switches were connected to external circuitry which switched the pump on when the low level switch was activated and off when the high level switch was activated by the magnet in the float. A stilling tube was fitted around the level control unit to provide a calm environment for the float and to prevent any splashes which occured from entering and sinking the float.

Three connections for a pressure gauge, liquid inlet and vapour outlet were made into the top of the vessel. The liquid inlet was fitted with a down pipe, which extended to below the minimum liquid level in the vessel. This minimised the splashing caused by the liquid being pumped into the vessel. A thermowell to accomodate the thermostat sensor bulb and a connection for the pressure switch were also provided in the top of the vessel.

Plate 3.3 shows a photograph of the generator with the insulation removed. Clearly visible in the photograph is the absorbent inlet, vapour outlet, pressure switch, thermostat, sight glass, drain and absorbent outlet.

FIG 3.4 GENERATOR FLOW DIAGRAM

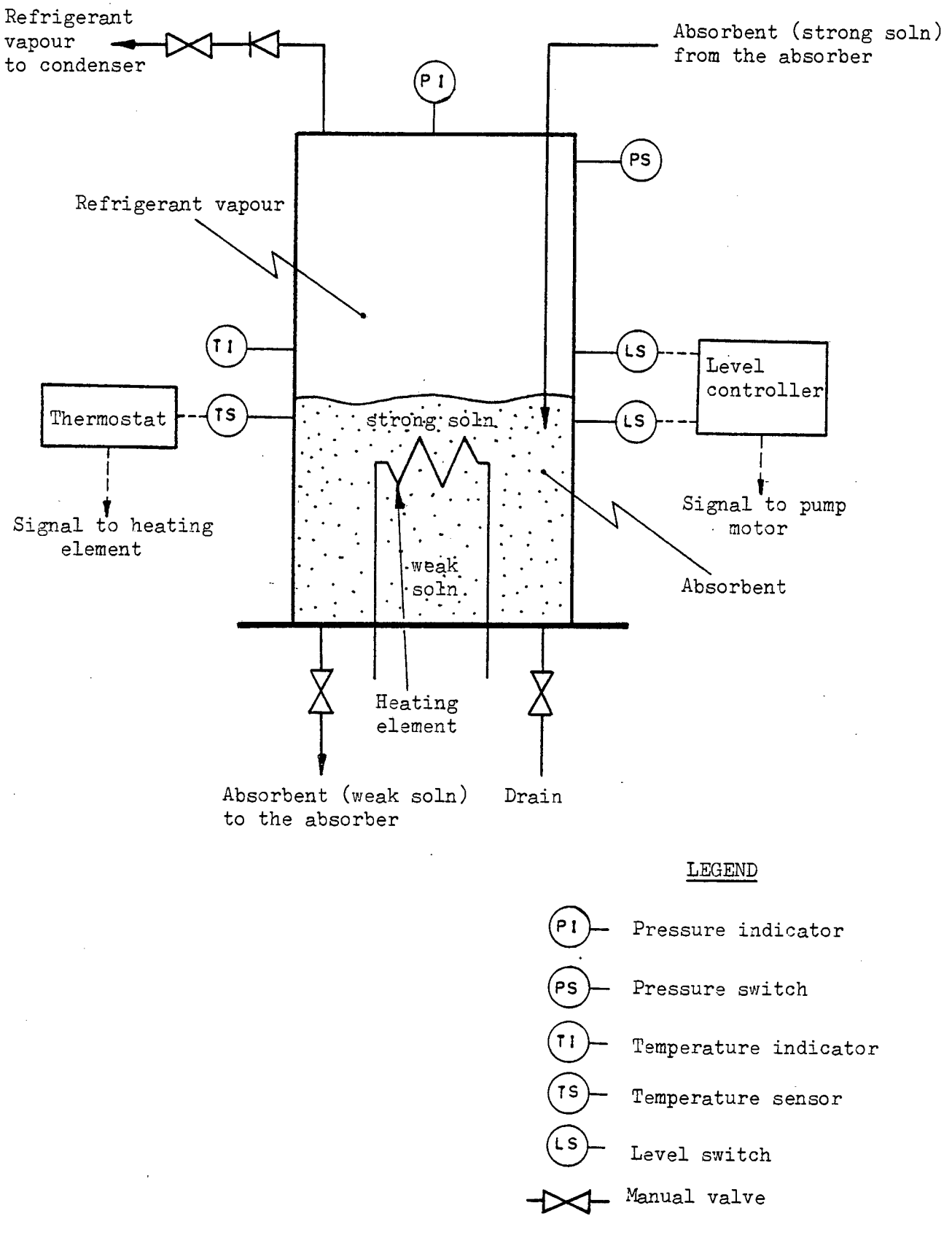


FIG 3.5 SECTION THROUGH THE GENERATOR

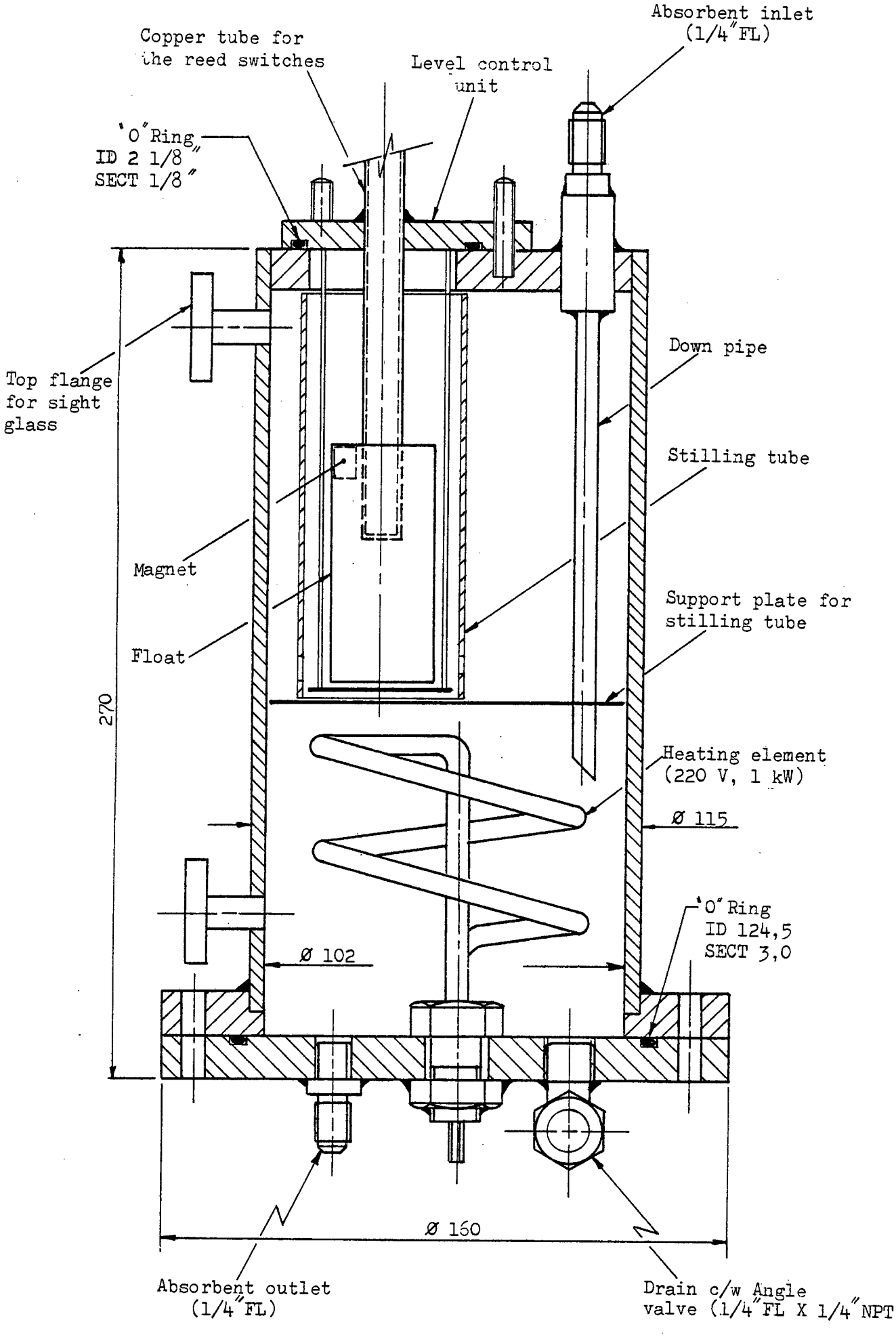
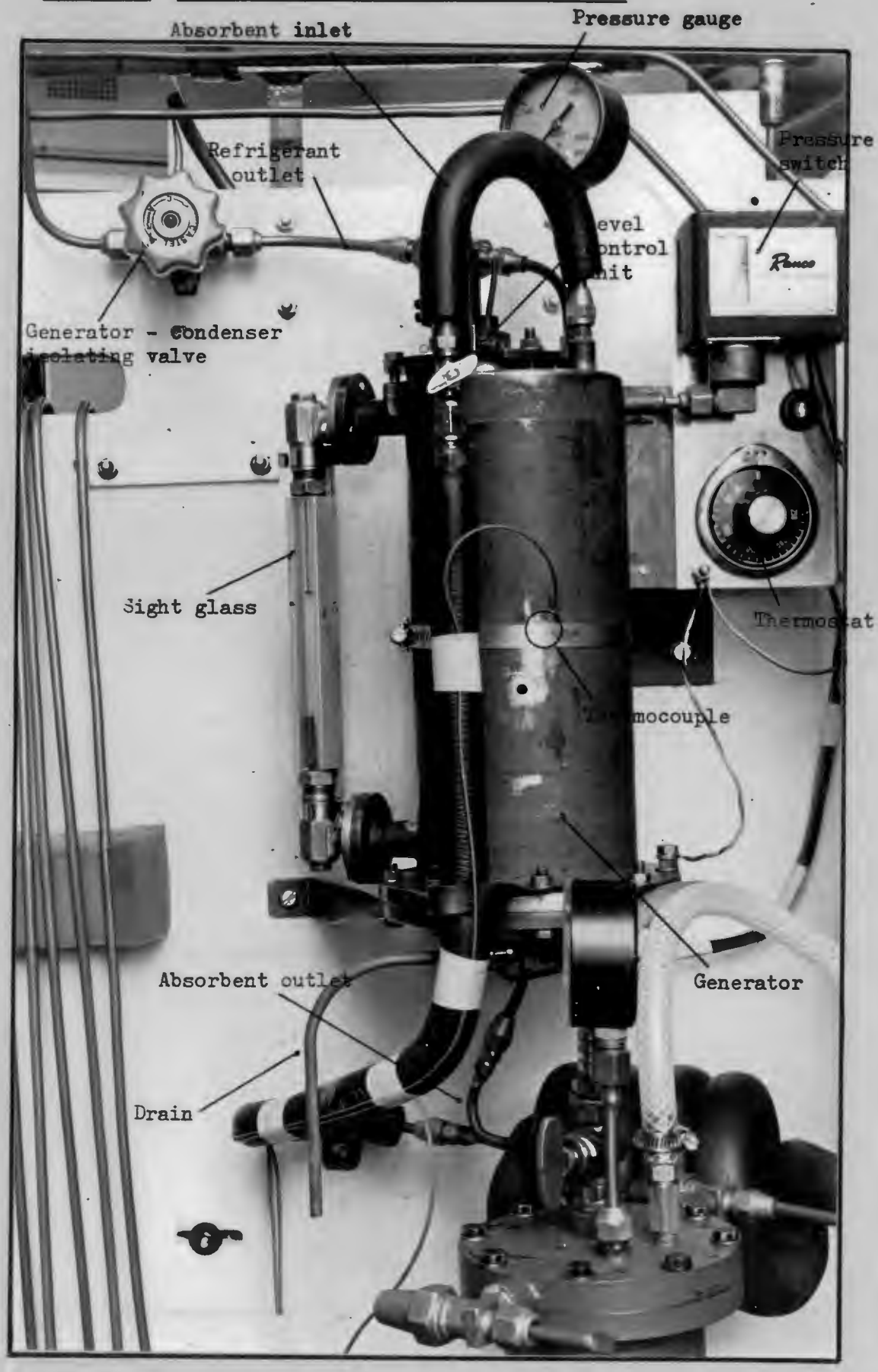


PLATE 3.3 GENERATOR WITH THE INSULATION REMOVED



3.4 THE ABSORBER

Figure 3.6 shows the flow diagram of the absorber. Warm absorbent (dimethyl ether - tetraethylene glycol) lean in absorbed refrigerant (difluoro chluoro methane) was introduced through a pressure reducing valve into the top of the absorber. This weak solution was distributed over the whole area of the vessel by a perforated distribution plate and then allowed to cascade over the cooling coils located in the upper section of the vessel. Cool weak absorbent passed off the bottom of the coils and collected in the reservoir at the bottom of the vessel where refrigerant vapour from the expansion column was passed through. In order to improve the absorption rate, which depended on the contact surface area between the fluids, the refrigerant vapour entering the absorber was passed through a sintered glass disc which broke the vapour stream into a large number of fine bubbles. This effectively increased the surface area of the vapour many fold and the absorption rate was improved as a result. Any of the vapour which was not absorbed during its passage through the absorbent reservoir rose into the upper section of the absorber which contained the cooling coils. In this section it became exposed to a large surface area of absorbent covering the cooling coils which enabled further absorption to occur. Non condensable gasses such as air which entered the absorber and which were not absorbed, collected at the top of the absorber from where they were purged from the system using the air bleed valve. Absorbent rich in absorbed refrigerant (strong solution) was extracted out of the absorbent reservoir. Since the strong solution was of lower density than the weak solution it would tend to be located at the top of the liquid in the reservoir. To account for this the absorbent was extracted from the upper half of the liquid in the reservoir by means of an extension piece fitted to the absorbent outlet port. This extension piece was made conical in shape with a larger area at the top. This reduced the velocity of the absorbent entering the

outlet and minimised the chance of unabsorbed bubbles of refrigerant vapour being sucked down the outlet and into the pump.

Figure 3.7 shows a cross section of the absorber vessel which consisted of an upper section containing the cooling coils and a lower section which formed the absorbent reservoir. The upper section of the absorber was constructed from a steel pipe which was flanged at the ends and closed at the top with a blank flange. A cooling coil was constructed from 16 metres of 3/16 inch copper tube wrapped in 3 layers (spaced 2,5 mm) around a central 3/4 inch copper pipe which was attached to the top flange. The ends of the coil were passed through the flange and connections for the water supply pipes were provided. Also included into the top flange was an air bleed port complete with a shut off valve and an absorbent inlet port complete with a pressure reducing valve.

The lower reservoir section of the absorber was constructed from a thick walled mild steel cylinder which was fitted with two portholes enabling the observation of the vapour inlets at the bottom of the absorber. This reservoir section was clamped between the bottom flange and the lower flange of the upper section where it was sealed with "O" rings. This section was used as a replacement piece for a glass cylinder which was originally used for the lower section but which was unable to withstand the pressure involved.

The bottom flange made provision for five vapour inlets, an absorbent outlet and a drain. The absorbent outlet complete with the screwed extension piece and a valve was located centrally in the reservoir. The five vapour inlet ports were then evenly spaced around it. A retaining plate which contained five sintered glass discs (dia. 20mm, porosity 4) was secured in position over the inlet ports such that each port was covered by a disc. The bottom flange was also

fitted with a drain.

Plate 3.4 shows a photograph of the absorber. Clearly visible in the picture are the viewing portholes, vapour inlets, pressure gauge, air bleed valve and absorbent outlet.

FIG 3.6 ABSORBER FLOW DIAGRAM

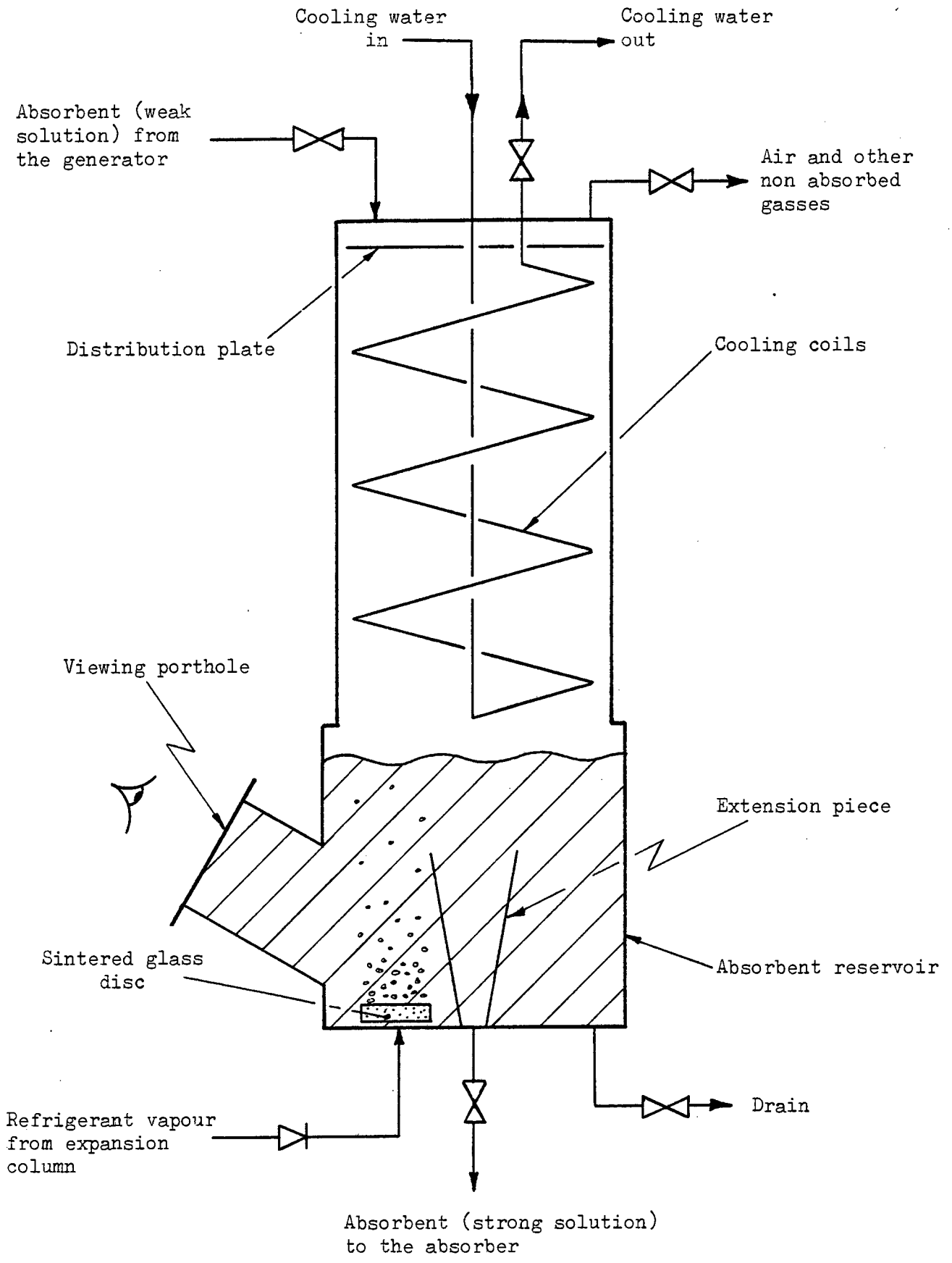


FIG 3.7 SECTION THROUGH THE ABSORBER

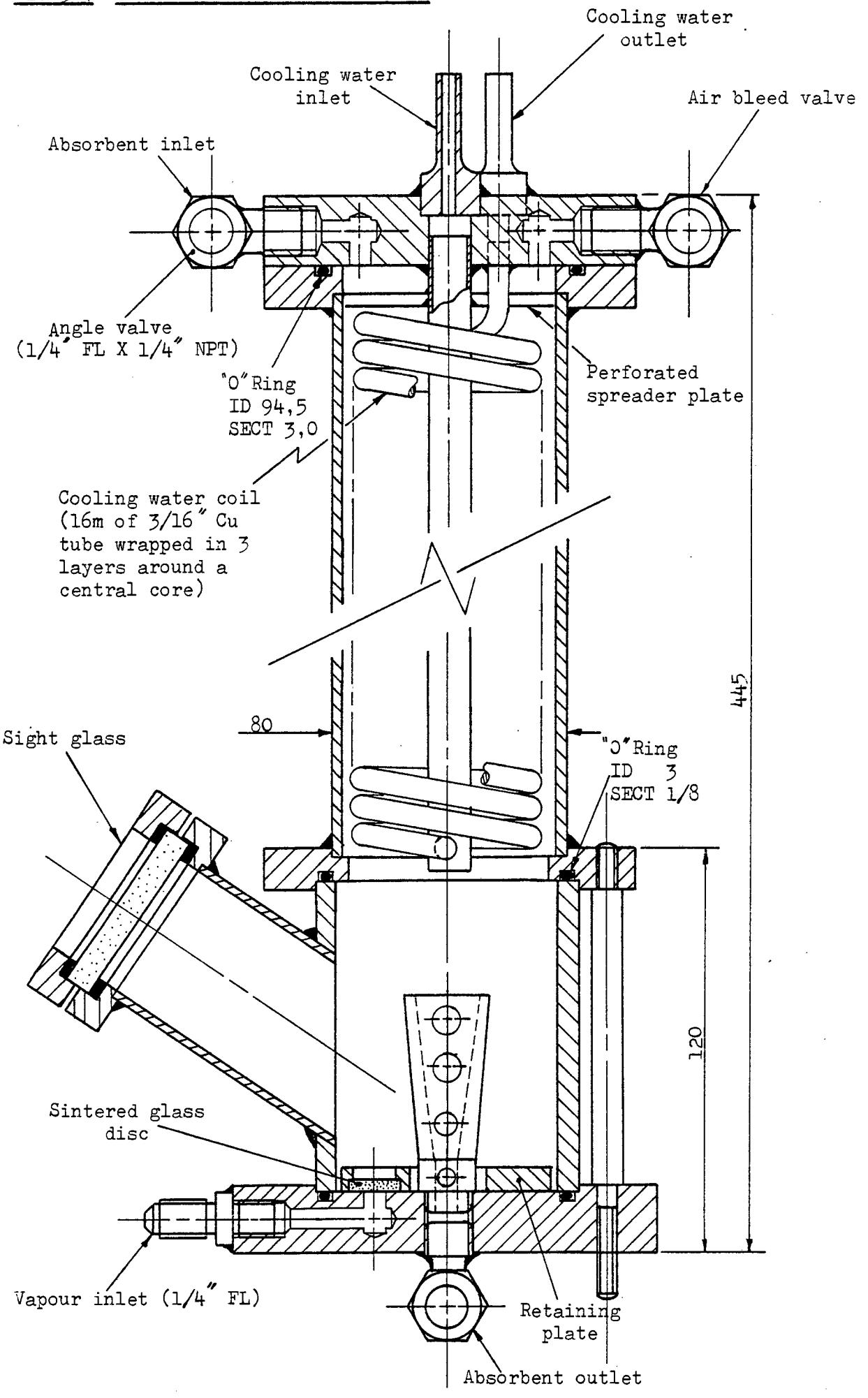
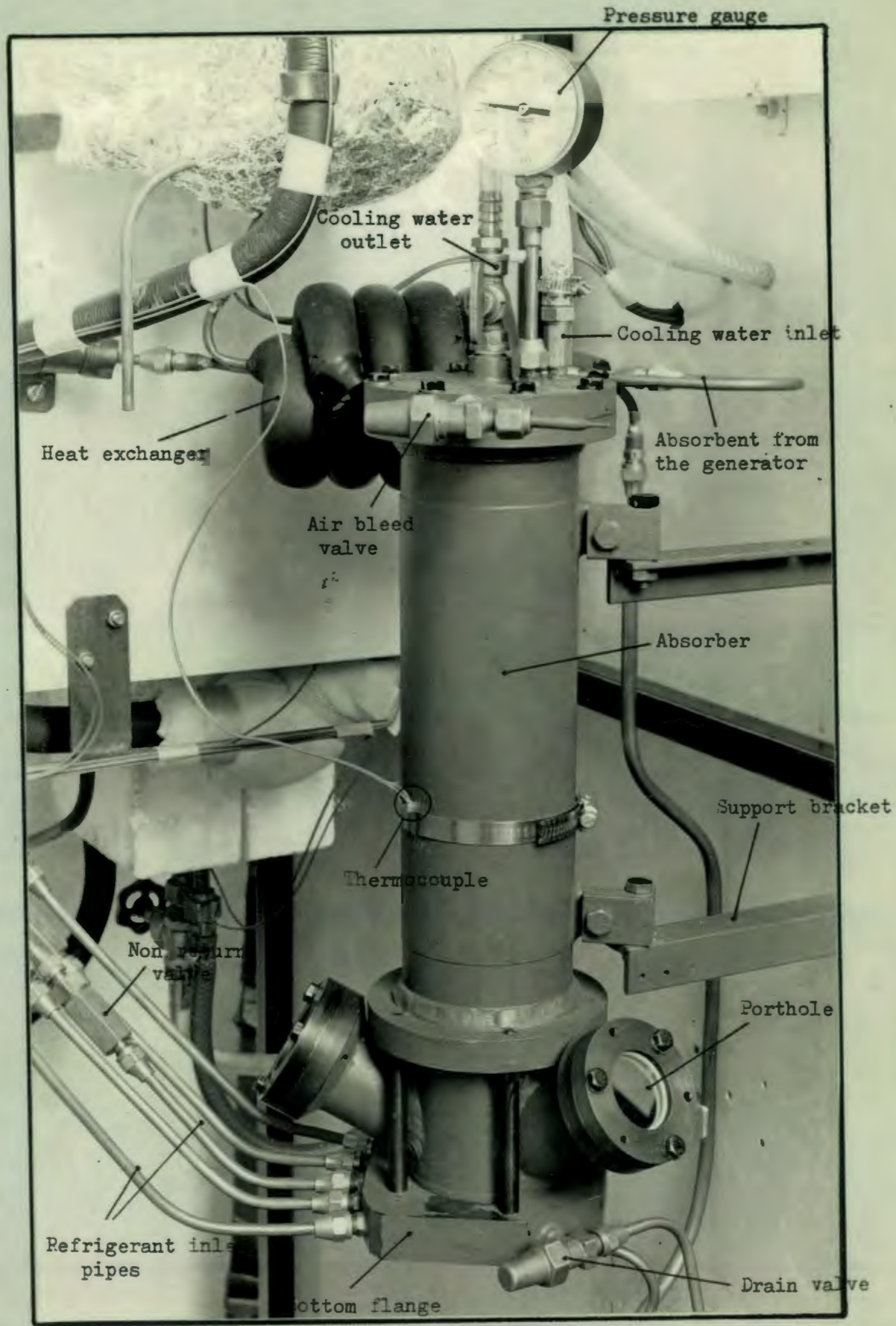


PLATE 3.4 THE ABSORBER



3.5 THE EVAPORATOR AND CONDENSER

The evaporator was of shell and tube design with the refrigerant on the shell side and the heating water on the tube side. The vessel was constructed of a copper cylinder (approximately 300mm long and 80mm diameter) with a single U tube coil which passed through the lower end plate. The evaporator vessel was orientated vertically with the refrigerant liquid inlet at the bottom and the vapour outlet at the top. The evaporator was positioned relative to the expansion column so that it operated approximatly half full of liquid refrigerant which then evaporated from the surface to produce the refrigerating effect. Heat was supplied to the evaporator by passing water at ambient temperature through the internal coil. Thermocouples attached to the water inlet and outlet recorded the temperatures of the water entering and leaving the evaporator. The heating water mass flow rate was determined by filling a container for a measured time and then weighing the contents.

The condenser was similarly of shell and tube design and like the evaporator the refrigerant was on the shell side and the cooling water on the tube side. The vessel was constructed of a copper cylinder approximately 450mm long and 60mm diameter. The condenser was orientated horizontally with the vapour inlet uppermost and the liquid outlet at the bottom on the opposite end of the vessel to the inlet. Heat was removed from the condenser by water at ambient temperature which was passed through the cooling coils. Thermocouples attached to the water inlet and outlet were used to measure the temperature of the water entering and leaving the condenser. The water flow rate was measured as for the evaporator.

3.6 THE PUMP UNIT

The pump unit consisted of a modified positive displacement hydraulic pump driven at reduced speed by a quarter horsepower electric motor. A vee belt drive was used to transmit the power and reduce the speed. The unit was constructed onto a steel base plate which was placed across the bottom of the frame and was complete with a charging reservoir, 3 way charging valve and strainer.

Modifications which were made to the pump to make it suitable for use with the chemically aggressive absorbent included :

i) The replacement of the original piston and cup seal which were not resistant to the absorbent with a PTFE piston. The replacement piston was also redesigned to reduce the clearance volume and thus reduce the loss of pumping capacity which occured because of dissolved vapour being released from the solution.

ii) The halving of the pump capacity by adapting the pump so that only one of the two pistons was in operation.

iii) The replacement of the steel valve discs with nylon discs which were found to provide better valve operation at the low speed at which the pump was used.

iv) The replacement of the original plastic valve spring retainers which were not resistant to the absorbent with similar ones manufactured from brass

3.7 THE COMPRESSOR UNIT

The compressor unit was of standard purchase and consisted of a two cylinder Bitzer Type "O" refrigeration compressor belt driven by a half horsepower electric motor. The compressor was complete with 3 way charging and evacuating valves on the inlet and outlet ports. An oil separator was fitted to the compressor discharge which removed compressor oil entrained in the refrigerant vapour and returned it to the crankcase. This was necessary to minimise oil contamination in the rest of the system and to ensure that the crankcase oil level was maintained.

The electric motor was mounted on trunions and equiped with a torque arm and balance weight which enabled the mechanical power produced by the motor to be measured.

3.8 INSTRUMENTATION AND DATA MEASUREMENT METHODS

System temperatures at the locations shown in Figure 3.1 and listed in appendix C were measured using Nickel - Chromium / Nickel - Aluminium thermocouples. A multi - selector switch was used to connect the thermocouples to a digital voltmeter which was used to measure the emf. An ice bath was provided for the reference junction so that the temperatures were measured relative to zero degrees celcius.

The system pressures were measured using bourdon tube pressure gauges of standard purchase located as shown on the flow diagram of Figure 3.1.

The electric energy consumed by the pump motor, compressor motor and generator heating element was measured using standard domestic kilowatthour (kwh) meters. Because the heater and the absorbent pump operated intermittently readings of the instantaneous power drawn by these items

were meaningless. Therefore the total energy consumed was measured for a known period of time and the average power consumption calculated. Each of the above items were fitted with a kwh meter from which readings were taken before and after a measured time period. These readings were then subtracted to determine the number of units of electricity consumed during the period. Division of the energy consumed by the time gave the average power consumption. To allow for short term power fluctuations the test runs were conducted over a period of one hour. Although the instantaneous compressor power was nominally equal to the average power (due to its continuous steady operation) the above procedure was never the less used for the compressor as it eliminated the effects of short term power fluctuations.

The compressor mechanical power was determined from the motor torque and speed which were measured using the motor torque arm and a hand held tachometer respectively. The pump mechanical power was not measured because the short intermittent periods for which it operated made it impossible to balance the pump motor torque arm. In any event the pump power was negligably small so its measurement was not critical.

The evaporator, condenser and absorber water flow rates were determined using a container, scale and stop watch. To measure the flow rate the container was filled with the water for a known time. The contents of the container were then weighed and the flow rate calculated by dividing the mass of water collected by the collection time.

3.9 EXPERIMENTAL METHODOLOGY

The experimental apparatus was constructed such that it could be operated both as a multiple expansion - absorption cycle and as a single expansion and compression cycle. To enable the two cycles to be compared easily the apparatus was first run in multiple expansion - absorption mode and then in single expansion mode between the same evaporating and condensing pressures.

In multiple expansion - absorption mode the apparatus was started and the expansion and flash gas throttling valves adjusted until suitable expansion pressures were achieved. A period of about one hour was then allowed for the apparatus to stabilise. Once steady state was achieved a set of readings was obtained from the kwh meters. The apparatus was then allowed to operate for one hour at constant conditions during which time the cooling and heating water flow rates were determined as described in section 3.8. At the end of the test period the kwh meter readings and the time were again recorded. The system temperatures were then recorded using the multi - selector switch to select each thermocouple location in turn. Finally the system pressures were recorded from the pressure gauges. The apparatus was then changed to single expansion mode.

In single expansion mode the expansion valve (manually operated needle valve) and condenser cooling water flow rate were adjusted until the evaporating and condensing pressures were the same as those used for multiple expansion operation. The system was then allowed to stabalise for about one hour. Once steady state was achieved a set of readings was obtained by recording the compressor kwh meter reading and the time. The apparatus was then allowed to operate at constant conditions for about half an hour during which time the evaporator and condenser water flow rates

were measured. At the end of the time period the kwh meter reading and the time were again recorded. The sytem temperatures and pressures were then recorded as before concluding the test.

CHAPTER 4

4.1 RESULTS

TABLE 4.1 SUMMARY OF THE RESULTS OF A THEORETICAL ANALYSIS OF THE PERFORMANCE OF A MULTIPLE EXPANSION VAPOUR COMPRESSION REFRIGERATION CYCLE

CYCLE TYPE	THEORETICAL COP	IMPROVEMENT OVER SINGLE STAGE CYCLE (%)	IMPROVEMENT OVER TWO STAGE INTER - COOLED CYCLE (%)
Carnot	5,84	---	---
Single Stage V.C.	4.60	---	---
Two Stage V.C. with Intercooling	4,83	5,0	---
Multiple Exp & Comp (5 Stages)	5,20	13,0	7,6
Multiple Exp & Comp (50 Stages)	5,33	16,0	10,5
Multiple Exp & Comp (100 Stages)	5,34	16,1	10,6

Refrigerant = R22

Condensing Temp = 35 ^{o}C Condensing Press = 13,1 bar

Evaporating Temp = -10 ^{o}C Evaporating Press = 3,5 bar

FIG 4.1 BAR CHART OF THEORETICAL COEFFICIENTS OF PERFORMANCE FOR VARIOUS REFRIGERATION CYCLES

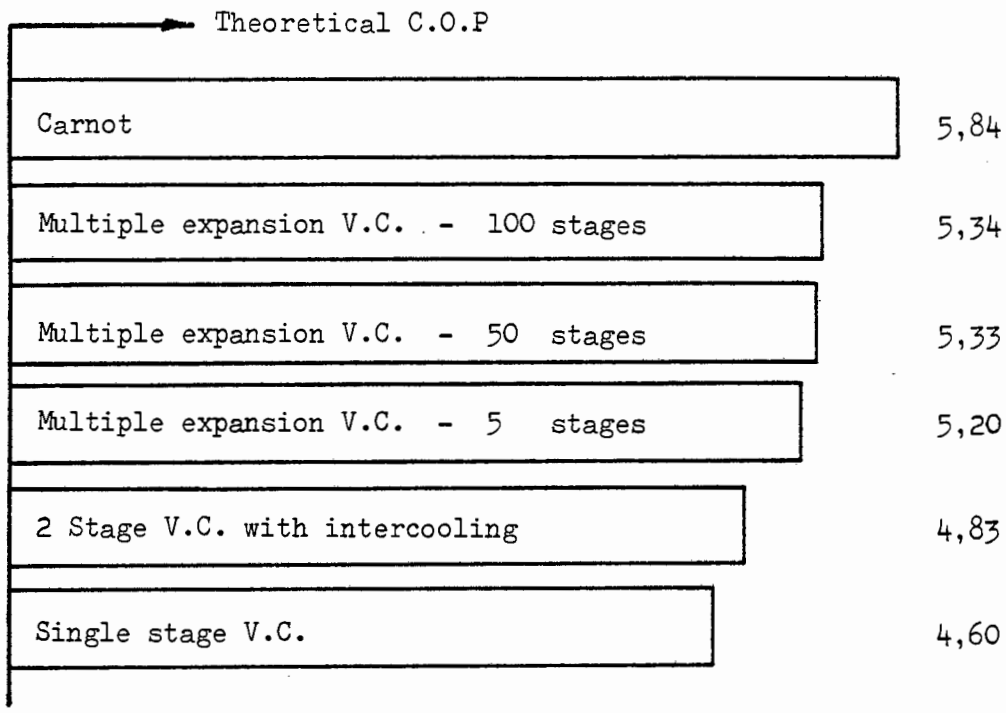


FIG 4.2 BAR CHART SHOWING THEORETICAL IMPROVEMENTS IN PERFORMANCE (%) OBTAINABLE WITH MULTIPLE EXPANSION V.C. CYCLES

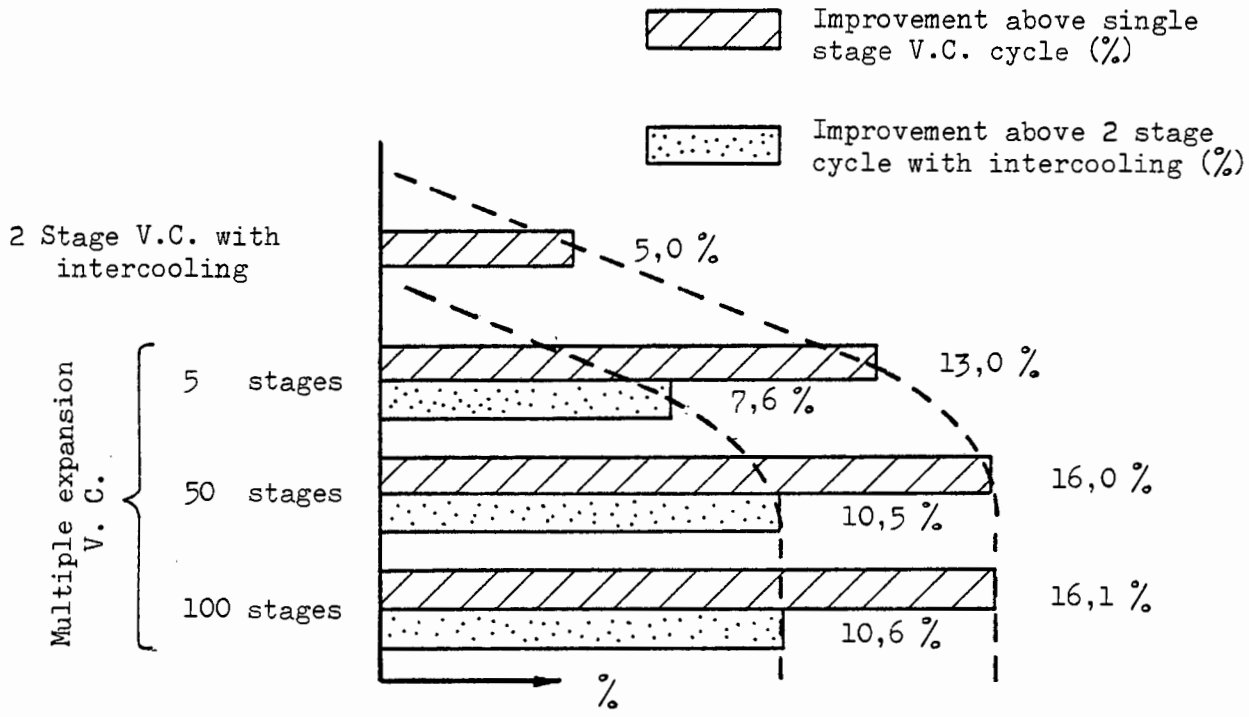


TABLE 4.2 SUMMARY OF EXPERIMENTAL RESULTS OBTAINED WITH THE MULTIPLE EXPANSION - ABSORPTION APPARATUS

	DATA SET NO.	1	2	3

		DATA SHEET NO.	2A	3A	4A
	Evaporating Temp	(deg C)	-15,31	-18,77	-19,51
M	Condensing Temp	(deg C)	21,32	20,39	21,32
U	Generator Temp	(deg C)	99,51	117,04	117,78
L	Temp of Exp Stage 1	(deg C)	14,81	14,57	15,56
	Stage 2		11,85	9,38	9,14
E	Stage 3		2,22	0,00	0,99
X	Stage 4		- 7,65	-10,86	- 7,90
P	Temp of Exp Stage 5	(deg C)	-17,78	-18,27	-18,27
	Refrigerating Effect	(W)	238,17	255,06	208,46
A	Comp Mech Power	(W)	194,26	191,20	189,44
N	Comp Elec Power	(W)	302,00	319,00	317,00
D	Ave Heater Power	(W)	77,00	85,00	88,00
	Ave Pump Power	(W)	12,00	12,00	10,00
A	COP (comp elec + heater + pump elec)		0,609	0,613	0,502
B	COP (comp mech + heater + pump elec)		0,841	0,885	0,725
S	COP (comp mech + pump elec)		1,155	1,255	1,045
	COP (comp elec + pump elec)		0,759	0,771	0,638

		DATA SHEET NO.	2B	3B	4B
O	Evaporating Temp	(deg C)	-15,31	-18,77	-19,75
N	Condensing Temp	(deg C)	21,32	21,11	21,32
E	Refrigerating Effect	(W)	122,81	217,17	194,05
	Comp Mech Power	(W)	189,99	194,39	189,44
E	Comp Elec Power	(W)	310,00	317,33	296,00
X	COP (comp elec)		0,396	0,684	0,656
P	COP (comp mech)		0,646	1,117	1,024

4.2 DISCUSSION

Table 4.1 shows a summary of the results obtained from a theoretical analysis of the multiple expansion vapour compression refrigeration cycle. Shown are the theoretical COP's which were calculated for three multiple expansion cycles (with different numbers of expansion stages), a Carnot cycle, a single stage V.C. cycle and a two stage V.C. cycle with intercooling all operating between an evaporating temperature of -10 ^{o}C and a condensing temperature of 35 ^{o}C and using Refrigerant 22.

Figure 4.1 shows the theoretical COP's represented graphically. From this bar chart it can be seen that the multiple expansion cycles give better theoretical performances and come closer to the performance of the Carnot cycle than single and two stage intercooled cycles. It can also be seen that the theoretical performance of the multiple expansion cycle increases as the number of expansion stages increases. Figure 4.2 shows graphically the percentage improvements in performance of the multiple expansion cycles compared to the single and two stage intercooled cycle. From this bar chart it is clear that as the number of expansions increases the rate of improvement in performance decreases rapidly and there is a point at which further increases in the number of expansions produces no signifigant benefit. For example increasing from 50 to 100 stages, for the operating conditions considered, resulted in an increase in the COP from 5,33 to only 5,34. Such a minimal improvement in performance would clearly be too small ever to warrant the increased complexity and cost of the additional expansions and compressions required to achieve it. It can be concluded that any practical benefit to be had from multiple expansions would be obtained using only a relatively small number of expansion stages, say 10 or possibly 15.

The theoretical coefficients of performance were calculated assuming isentropic compressions of the refrigerant and did not allow for compressor friction losses. Thus the COP's which were calculated do not account for the increased friction losses which would occur in the extra compressors required for the multiple expansion vapour compression cycles. It is probable that using many small auxiliary compressors to compress the flash gas from each expansion would not in practice result in an improved performance as each of the compressors would suffer from disproportionately high friction losses compared to the useful work performed. In other words the additional friction losses in all of the extra compressors would probably be greater than the work saved by having multiple expansions. Use of a single auxillary flash gas compressor capable of compressing gas from various inlet pressures to a single discharge pressure would limit the extra compressor friction losses suffered by the system. However such a compressor is not available and would require considerable development if the concept was even possible. The most promising possibility for the compression of the flash gas in a multiple expansion vapour compression cycle would be the use of vapour jet ejectors. An ejector would be required for each stage and would operate off the main compressor discharge which would thus be divided into a number of paths corresponding to the number of stages. Since ejectors would be cheap and essentially friction free their use would overcome the problems of increased cost and friction losses resulting from the use of additional compressors for the flash gas. The use of ejectors in a multiple expansion cycles is an aspect which could be investigated further.

The coefficient of performance of the multiple expansion - absorption refrigeration cycle can, like other refrigeration cycles, be broadly defined as the quotient of the refrigerating effect over the energy supplied to achieve it. In simple terms the benefit gained over the price paid. The

multiple expansion - absorption cycle had three energy inputs to be considered in defining the coefficient of performance. These were the compressor work, the pump work and the generator heat input. In some industrial applications the heat input might be supplied from waste energy which would otherwise be rejected to the environment. In such a situation it could be argued that since the heat energy required is "freely" available to the system it can be legitimately excluded from the definition of the COP for the system for that application. Then comparison of the COP's of competing systems for that particular application give an indication of how much "new" energy must be supplied and therefore the energy costs to achieve the desired refrigerating effect using the different systems. The coefficient of performance of the multiple expansion - absorption system was therefore defined both with and without the heat input included. Also as the use of the compressor electrical power input in determining the COP would attribute the motor and drive inefficiencies (which were high for the small motors and belt drives used) to the refrigeration cycle the COP was also variously defined using electrical or mechanical power inputs. Thus four definitions of the COP of the experimental multiple expansion - absorption cycle were considered. These were as follows :

i) COP1 = R.E./Comp elec + Heater + Pump elec

ii) COP2 = R.E./Comp mech + Heater + Pump elec

iii) COP3 = R.E./Comp mech + Pump elec

iv) COP4 = R.E./Comp elec + Pump elec

Where

R.E. = Refrigerating effect
Comp elec = Compressor electrical power input
Comp mech = Compressor mechanical power input
Pump elec = Pump electrical power input

In absorption refrigeration cycles it is common to neglect the pump work when calculating the COP since it is small in comparison to the heat input. However as the average pump electrical power input was measured during the tests on the apparatus it was included in the calculation of the COP's.

The experimental apparatus was designed to enable it to be operated as a multiple expansion - absorption cycle and as a single stage V.C. cycle. This enabled the performance of the multiple expansion - absorption cycle to be compared easily to the performance of the single stage cycle. Thus the apparatus was first operated in the multiple expansion - absorption mode and then in the single stage V.C. mode. The same evaporating and condensing pressures were used in each mode.

Table 4.2 shows a summary of the experimental results which were obtained with the multiple expansion - absorption refrigeration apparatus.

Because the compressor was operated continuously at constant speed between essentially the same suction and discharge pressures the compressor power in both the single and multiple expansion modes was expected to remain constant. This was confirmed by the measured compressor power which as shown in Table 4.2 remained more or less constant for all of the tests performed.

In the multiple expansion mode the refrigerant entered the evaporator at a lower enthalpy than it did in the single

expansion mode. Since the mass flow rate of refrigerant through the evaporator was constant (due to the constant operating pressures and speed of the compressor) the refrigerating effect was expected to be greater in the multiple expansion mode than in the single expansion mode. This was confirmed by the results in Table 4.2 which show that the refrigerating effect was of the order of 10% greater (data sets 2 and 3) in multiple expansion mode than in single expansion mode. The wide range of values recorded was due to the evaporator heating arrangements which were found to be unsatisfactory. The single water heating tube in the evaporator had insufficient surface area and too large a diameter (low water velocity) for good heat transfer. This resulted in only a few degrees of cooling being achieved which, because the water inlet and outlet temperatures were measured separately and then subtracted to determine the temperature drop through the evaporator, resulted in poor accuracy in the equations. In order to improve the measurement of the refrigerating effect the evaporator would require a considerably longer heating tube of smaller diameter.

Considering data sets 2 and 3 in Table 4.2 in can be seen that the COP's obtained for the multiple expansion - absorption cycle, when all energy inputs were included, were slightly less than those obtained for the single stage V.C. cycle. For a rigorous analysis this is the correct way to define the COP of the multiple expansion - absorption cycle which must therefore be considered as inferior to the single stage V.C. cycle in any simple refrigeration application where electrical power is used as the driving force. This is made even more apparent when the increased complexity and cost of the multiple expansion - absorption system is considered. However when the generator heat input was neglected then the COP obtained for the multiple expansion - absorption cycle was slightly better than that obtained for the single stage V.C. cycle. The improvement measured was

however small and cannot be considered as conclusive due to the inadequacies and poor accuracy of the measurement of the refrigerating effect. The results do however indicate that some improvement in the COP is obtainable with the multiple expansion - absorption cycle if the generator heat input can be neglected for the application considered. It could of course be argued that if the generator heat input is to be neglected then a straight forward absorption cycle, which has only heat inputs which could all be neglected, would give the best performance. This argument is valid and in such a case where abundant waste heat is available the simple absorption cycle would be superior. It is clear that the multiple expansion - absorption cycle would not be suitable for simple refrigeration applications as it cannot compete either on the basis of cost or performance with the conventional V.C. and absorption systems. The cycle may however be suitable for some special applications. These might be where only limited waste heat, insufficient for a full absorption cycle, is available or where some minor refrigeration is also required at temperatures above the main evaporating temperature. In such an application some saturated liquid refrigerant would be drawn from the appropriate expansion stage to effect the refrigeration required at the intermediate temperatures. The absorption equipment would then handle refrigerant vapour from the intermediate temperature refrigeration as well as the flash gas.

The successful operation of the experimental apparatus showed that the concept of a multiple expansion - absorption refrigeration system was achievable in practice even if it did not provide any improvement in performance compared to conventional refrigeration systems. Some aspects of various items of the apparatus are discussed below.

The expansion column was found to be succesfull in expanding the refrigerant to the evaporating pressure in a number of

stages. Initially float valves were used in each chamber to control the liquid level. The float valves were however unsatisfactory and were ultimately found to be unnecessary. One of the major problems encountered with the float valves was the strength of the floats which were required to be small, withstand pressures of up to 15 bar and be able to exert sufficient nett buoyancy force to close the needle valves. It was found to be very difficult in practice to meet these criteria using closed floats. All of the floats considered were either unable to withstand the pressure and collapsed or else were too heavy for their volume and could not provide sufficient force to operate the needle valves effectively. Open floats constructed with a small pressure equalising orifice above the liquid level were also tried without success. Their failure was due to the establishment of the same equilibrium system of condensing and evaporating refrigerant inside the float as that which existed outside the float. This eventually caused the floats to sink as liquid accumulated in them. Ultimately it was however found that the level control valves were an unnecessary complication as the expansion column operated quite satisfactorily without them. It was found that during normal operation the expansion column was quite stable and the stage pressures remained constant for long periods. Tests of up to six hours were performed with no significant fluctuations in the stage pressures being observed. It was found that the stage pressures were determined by the amount of throttling in both the expansion valves and the flash gas bleed off valves, both of which had to be adjusted to effect any pressure changes. At first it would seem that difficulty would be experienced in ensuring that the correct amount of flash gas was bled off from each expansion stage. However since an equilibrium mixture of liquid and vapour existed in the chamber the flash gas flow rate was self regulating. If more vapour than the available flash gas was allowed to escape from the chamber (by opening the bleed valve) then the additional vapour would have to come from liquid

refrigerant evaporated in the chamber. Since the chambers were insulated this would result in sensible heat being removed from the refrigerant in the chamber and the temperature would drop. Since an equilibrium mixture existed in the chamber the reduction in temperature would result in a reduction of pressure. This reduction of pressure would mean that the incoming refrigerant would undergo a larger expansion, more flash gas would be formed and less refrigerant evaporated in the chamber. The pressure would therefore drop until the equilibrium was re-established and the rate of flash gas formation equalled the rate at which the vapour escaped from the chamber. In the event of less vapour being bled from the chamber than the rate of flash gas formation the opposite would occur and the chamber pressure would increase until the equilibrium was re-established. Therefore as long as liquid and vapour were present in the chambers and the temperatures and pressures remained constant the rate of flash gas removal equalled the rate of flash gas formation. Due to the large mass of the expansion column a considerable thermal inertia was present which caused the stage pressures to respond slowly (several minutes) to adjustments of the expansion and flash gas bleed valves. As a result pressure stability was excellent under normal conditions.

Pressure instability arose when moisture was present in the refrigerant. As the temperature of the refrigerant fell the solubility of water in the refrigerant decreased and some of the absorbed moisture was released. This moisture froze in the expansion valves preventing the flow of refrigerant and causing the expansion chamber pressures to become unstable. It was found that once the expansion valves had frozen it was very difficult to thaw them. This was mainly because as the moisture froze it encapsulated small "pockets" of refrigerant to form a "sponge like" solid plug consisting of ice and trapped liquid refrigerant. As the ice plug started to melt some of the entrapped refrigerant was released and

immediately evaporated and refroze the water. This meant that unless heat was supplied to the blocked expansion valve it took a very long time for the ice plug to melt. It became critical to ensure that the refrigerant was moisture free. A silica gel drier, which subsequently maintained a satisfactory degree of dryness in the system, was therefore fitted into the liquid line.

The elaborate construction of the expansion column was due to the initial requirement for level control valves in each chamber. This meant that the chambers had to be reasonably large to accomodate the floats and be of such a design as to allow access to the inside of the chambers. Having shown that level control in the chambers was unnecessary the chambers could be made much smaller and designed to be permenantly closed. This would make the expansion column considerably more compact, easier to manufacture and cheaper.

The absorption related equipment was found in general to give satisfactory performance although some difficulties were experienced initially. These difficulties were mainly due to the chemically aggressive nature of the absorbent which attacked most common sealing materials. PTFE and Nylon which were resistant to attack from the absorbent were eventually used for most of the sealing applications.

The generator performed adequately in all respects and was operated successfully at various temperatures between 100 and 120 degrees Celcius. Better operation was obtained at the higher temperatures although clearly a temperature of 120 degrees Celcius would be a little high for possible industrial applications where waste heat would usually be available as flash steam at 1 bar and 100 degrees Celcius. A greater rate of circulation of the absorbent was required for the lower temperature operation. The open top float used in the generator level control unit gave satisfactory

performance although it was essential to have the stilling tube around the float to prevent its inadvertent sinkage. It was also necessary to ensure that the generator was not flooded at shut down as this resulted in the float being filled with liquid. To prevent flooding the generator was isolated by closing all of its valves at shut down.

The absorber was in general satisfactory although some cooling coils in its reservoir might have been an improvement. The use of sintered glass discs to increase the area of the refrigerant vapour exposed to the absorbent proved to be successful and together with the high affinity of the absorbent for the refrigerant no problems in absorbing the flash gas were encountered. The periodic build up of pressure in the absorber was due to the accumulation of non condensable gasses. The pressure build up was relieved from time to time by manually purging these gasses from the air bleed valve on the top of the absorber.

The regenerative heat exchanger used to preheat the absorbent being pumped to the generator was not entirely effective. This was mainly due to the very intermittent operation of the pump and the low flow rate of absorbent returning to the absorber. This meant that for most of the time there was no flow of cool absorbent from the absorber through the heat exchanger. Hot absorbent flowing through the heat exchanger from the generator was therefore not cooled when the pump was inoperative. When the pump was in operation its flow rate was so much greater than the rate at which absorbent returned to the absorber that no effective heat exchange occured. It was clear that in order for such a regenerative heat exchanger to be useful the flow of absorbent from the absorber to the generator would have to be continuous and at the same rate as the flow from the generator to the absorber. This requires a small capacity continuously operating pump rather than the high capacity intermittently operating pump which was used.

No rigorous evaluation of the refrigerant - absorbent combination was attempted due to the lack of any thermodynamic data on the absorbent, the fact that such an evaluation was beyond the scope of the project and the fact that such evaluations have been made by previous workers. However from a practical point of view the combination performed well. The main disadvantages of the absorbent which were evident were its extremely high cost and its chemical agressiveness to most sealing materials. The main advantages of the absorbent were seen to be its high affinity for Refrigerant 22 and its extremely high boiling point which meant that absorbent carry over from the generator into the condenser was not a problem. A further advantage was the fact that mild steel, copper, brass and aluminium could be safely used with the absorbent.

4.3 CONCLUSIONS

From the tests which were performed on the apparatus and the results obtained it can be concluded that :

i) Although the multiple expansion - absorption cycle worked satisfactorily its COP, when all the energy inputs were included, was found to be lower that that of a single stage V.C. cycle. Only when the generator heat input was neglected did the cycle have a better performance than that of the single stage V.C. cycle. The multiple expansion - absorption cycle would therefore not be suitable for simple refrigeration applications and would only find possible application in special circumstances. The low absolute values of the COP's obtained were due to the small scale of the apparatus which meant that friction and other losses were disproportionately high compared to the useful work performed.

ii) The tests which were performed showed that the concept of expanding the refrigerant in a number of stages using an expansion column constructed of a series of flash chambers was achievable in practice. It was found that the expansion column exhibited good stability and that no liquid level control mechanisms were necessary. The rate of flash gas escape from the chamber was found to be self regulating and no pressure fluctuations occured. The results showed that, as predicted by theory, the liquid refrigerant leaving the last stage of the expansion column had a lower enthalpy than that produced in a single expansion. Due to the inadequacies of the measurement technique it was not possible to determine a reliable figure for the increase in the refrigerating effect achieved although an approximately 10% increase was observed when using the expansion column. The success obtained with the expansion column

opens the way for its miniaturisation and the investigation of a practical (low friction) means of compressing the flash gas from each expansion. This would enable the production of a true multiple expansion vapour compression system with the potential for achieving the increased performance indicated by the theoretical analysis which was performed.

iii) Although no rigorous evaluation of the absorbent - refrigerant combination was performed, from a practical point of view, the combination was found to be satisfactory - the main disadvantage of the absorbent being its high cost.

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APPENDIX A

THEORY OF REFRIGERATION CYCLES

A.1 THE CARNOT REFRIGERATION CYCLE

The Carnot refrigeration cycle (reversed Carnot heat engine) is an ideal cycle which has the maximum possible performance of any refrigeration cycle operating between the same temperatures [1,6]. The cycle consists of two isothermal heat transfers, an isentropic expansion, an isentropic compression and an isothermal compression. Figure A.1 illustrates the concept of a reversed Carnot heat engine. Heat is absorbed from the cold side at a temperature T_{cold} and delivered to the hot side at a temperature T_{hot}. An amount of work W must be supplied to achieve the heat flow against the temperature gradient. Although the Carnot cycle cannot be implemented in practice, due to the impossibility of achieving the partly adiabatic and partly isothermal compression required, the cycle is valuable as a basis of comparison for other refrigeration cycles.

Figure A.2 shows the Carnot refrigeration cycle on a temperature - entropy (T-s) diagram. Refering to this diagram, saturated vapour from the evaporator at condition (1) is compressed isentropically and adiabatically to condition (2). This condition is superheated vapour at some pressure between the evaporating and condensing pressures. This vapour is then compressed isothermally to condition (3) which is saturated vapour at the condensing pressure. Saturated vapour at condition (3) is condensed to condition (4) by means of an isothermal heat transfer during which heat is rejected to the heat sink. The resulting saturated liquid is then expanded isentropically to condition (5) which is an equilibrium mixture of saturated liquid and vapour at the evaporating pressure. During this expansion work must be extracted from the refrigerant. The saturated liquid from the expansion is evaporated in the evaporator to produce the refrigerating effect.

FIG. A.1 REVERSED CARNOT HEAT ENGINE

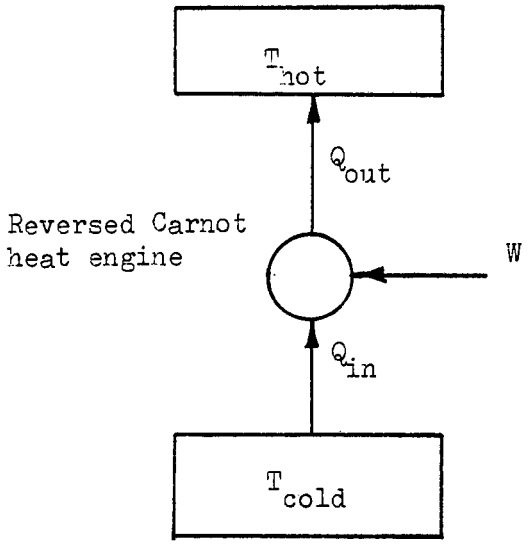
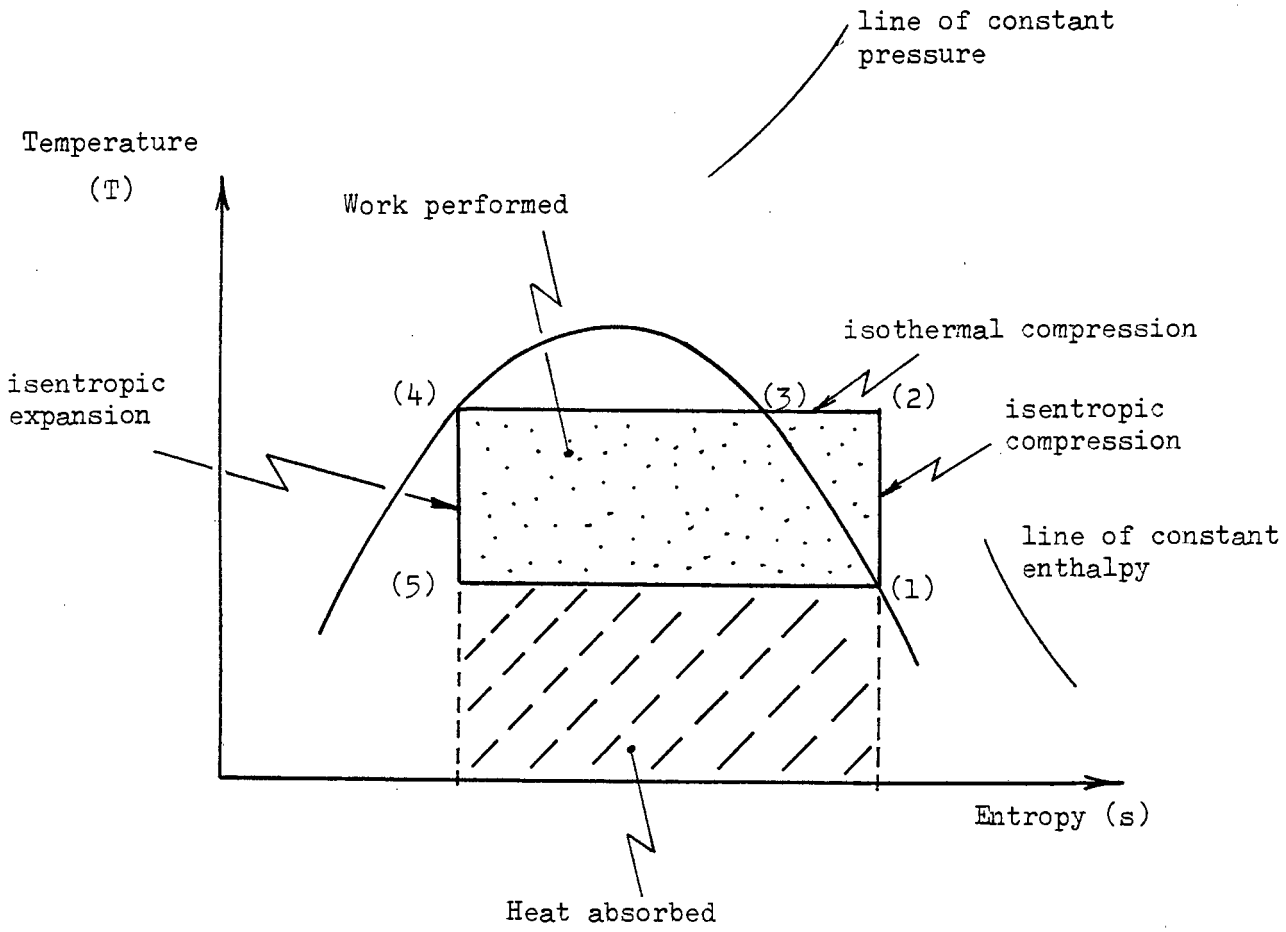


FIG. A.2 CARNOT REFRIGERATION CYCLE ON A TEMPERATURE - ENTROPY DIAGRAM



The coefficient of performance of the Carnot refrigeration cycle is given by the equation :

$$COP_{carnot} = Q_{evap}/W \qquad \text{---} \quad (A.1)$$

But

$$W = Q_{cond} - Q_{evap}$$

Thus

$$COP_{carnot} = Q_{evap}/(Q_{cond} - Q_{evap}) \qquad \text{---} \quad (A.2)$$

Where

COP_{carnot} = Coefficient of performance of the Carnot refrigeration cycle.
Q_{evap} = Heat absorbed in the evaporator (J/s).
Q_{cond} = Heat rejected in the condenser (J/s).
W = Work supllied to the reversed Carnot engine (J/s).

For isothermal heat transfers the rate of heat transfer is given by the equation :

$$Q = m \cdot T \cdot ds$$

Where

Q = Rate of heat transfer (J/s).
T = Temperature at which heat transfer occurs (K).
ds = Change in the entropy of the working fluid during the heat transfer (J/kg k).
m = Working fluid mass flow rate (kg/s).

Substituting this relationship into equation (A.2) gives :

$$COP_{carnot} = \frac{m \cdot T_{evap} \cdot (S_1 - S_5)}{[m \cdot T_{cond} \cdot (S_1 - S_5)] - [m \cdot T_{evap} \cdot (S_1 - S_5)]}$$

Therefore

$$COP_{carnot} = T_{evap} / (T_{cond} - T_{evap}) \quad --- \quad (A.3)$$

Where

T_{evap} = Evaporating temperature (K).
T_{cond} = Condensing temperature(K).
S_1 = Entropy at cycle point 1 (J/kg K).
S_5 = Entropy at cycle point 5 (J/kg K).
m = Working fluid mass flow rate (kg/s).

From equation A.3 it can be seen that a decrease in the evaporating temperature and or an increase in the difference between the evaporating and the condensing temperatures results in a decrease in the coefficient of performance of the Carnot cycle.

A.2 THE SINGLE STAGE VAPOUR COMPRESSION CYCLE

The single stage vapour compression (V.C.) cycle consists of an evaporator, a condenser, a compressor and an expansion device. Figure A.3 shows the basic V.C. cycle [5,6].

The refrigerating effect is produced in the evaporator by evaporating a suitable refrigerant at a low pressure and temperature. Heat is rejected from the cycle in the condenser which is usually cooled with air or water at ambient temperature. In order for this heat rejection to occur, the pressure of the refrigerant has to be raised until the saturation temperature is greater than the temperature of the cooling medium. In the V.C. cycle a compressor is used to compress the refrigerant vapour to the condensing pressure.

The single stage V.C. cycle can be represented on a pressure enthalpy diagram as shown in Figure A.4. Saturated vapour from the evaporator at condition (1) is compressed (assumed isentropically) from the evaporator pressure (Pevap) to the condenser pressure (Pcond). Superheated vapour leaving the compressor at condition (2) enters the condenser where it is first de-superheated and then condensed to form saturated liquid at condition (3). Using a constant enthalpy expansion device this saturated liquid at the condensing pressure is expanded to condition (4). During this expansion some of the liquid flashes to vapour. Condition (4) is therefore an equilibrium mixture of saturated liquid at condition (5) and saturated vapour at condition (1). The saturated liquid at condition (5) is evaporated in the evaporator to saturated vapour at condition (1). During this process the latent heat of evaporation of the refrigerant is absorbed and a refrigerating effect is produced.

FIG A.3 SINGLE STAGE VAPOUR COMPRESSION CYCLE

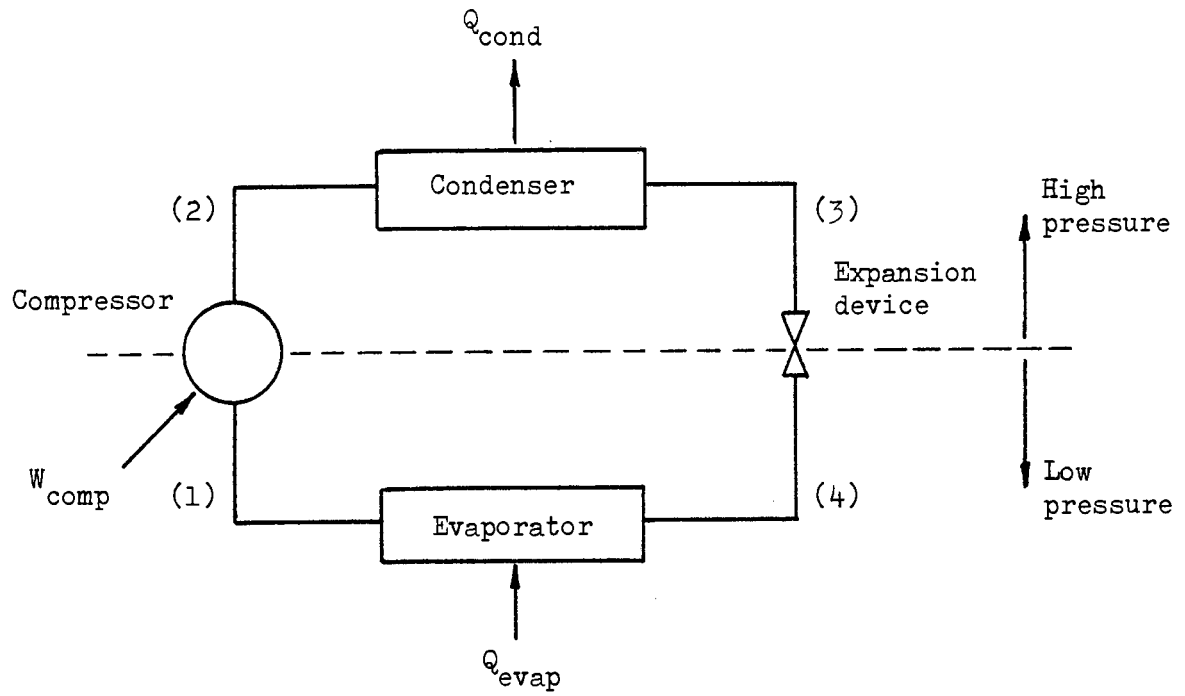
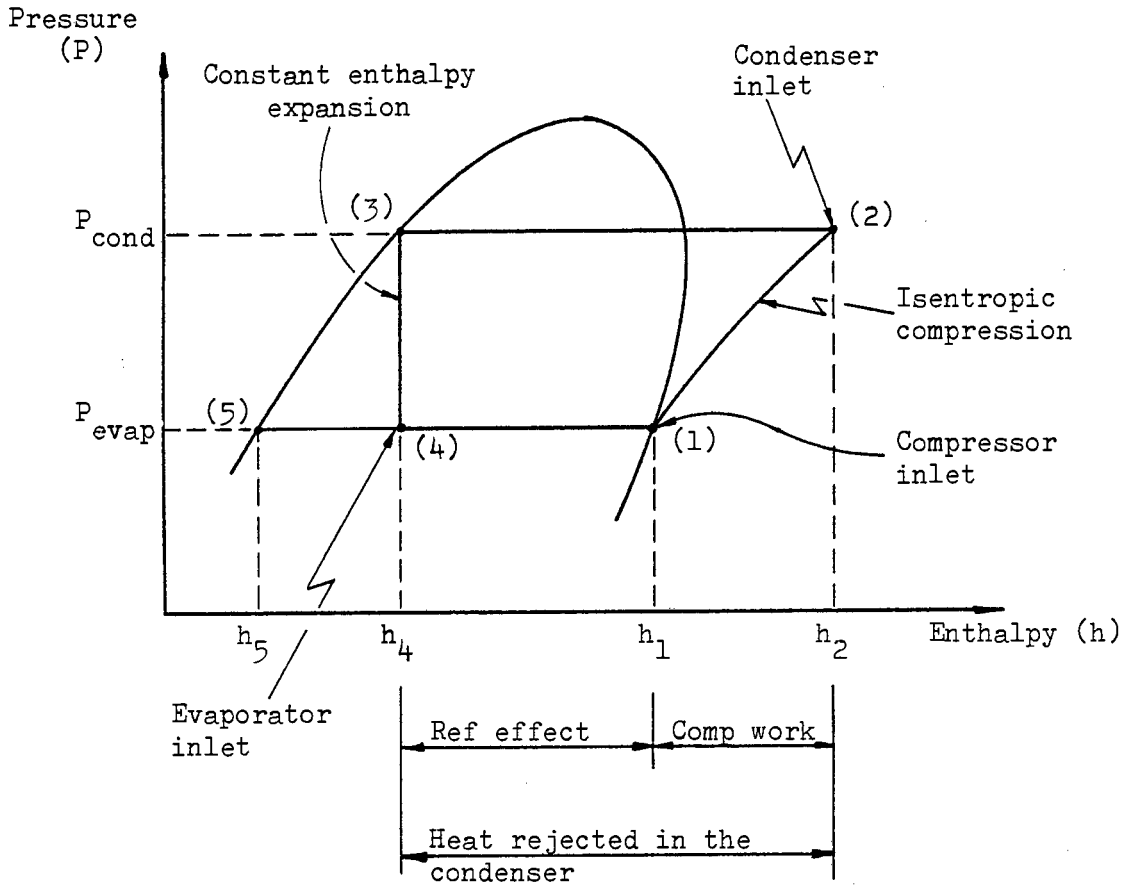


FIG A.4 SINGLE STAGE VAPOUR COMPRESSION CYCLE ON A PRESSURE - ENTHALPY DIAGRAM



The coefficient of performance of a single stage vapour compression cycle has been defined as the ratio of the heat absorbed in the evaporator to the work supplied to the compressor. The COP is therefore given by the equation :

$$COP_{vc} = Q_{evap}/W \qquad \text{---} \quad (A.4)$$

Where

COP_{vc} = Coefficient of performance of a single stage vapour compression cycle.
Q_{evap} = Heat absorbed in the evaporator (J/s).
W = Compressor work (J/s).

But

$$Q_{evap} = m_1 \cdot (h_1 - h_4)$$

$$W = m_1 \cdot (h_2 - h_1)$$

Where

h_i = Enthalpy at cycle point i (J/kg) (Refer to Fig. A.4).
m_i = Refrigerant mass flow rate at cycle point i (kg/s).
i = Cycle point number.

Thus

$$COP_{vc} = (h_1 - h_3)/(h_2 - h_1) \qquad \text{---} \quad (A.5)$$

A.3 COMPARISON OF THE V.C. CYCLE WITH THE CARNOT CYCLE

Figure A.5 shows a vapour compression and a Carnot refrigeration cycle superimposed on a temperature - entropy diagram.

The isentropic expansion of the Carnot cycle is replaced in the V.C. cycle with a constant enthalpy expansion. This results in a reduction in the refrigerating effect. A constant enthalpy expansion is normally used in real refrigerating systems because the work which can be extracted with an isentropic expansion is small and the additional equipment required for the expansion is not economically justifiable. Since the partly isentropic and partly isothermal compression required by the Carnot cycle is not achievable with real compressors the ideal V.C. cycle has a completely isentropic compression. As can be seen in Figure A.5 this results in a "superheat horn" on the V.C. cycle. As a result high compressor discharge temperatures occur and more compressor work has to be performed in the V.C. cycle than in the Carnot cycle. For these reasons the coefficient of performance of the V.C. cycle is always somewhat less than the COP of the Carnot cycle.

FIG A.5 CARNOT AND REAL REFRIGERATION CYCLES ON A TEMPERATURE - ENTROPY DIAGRAM

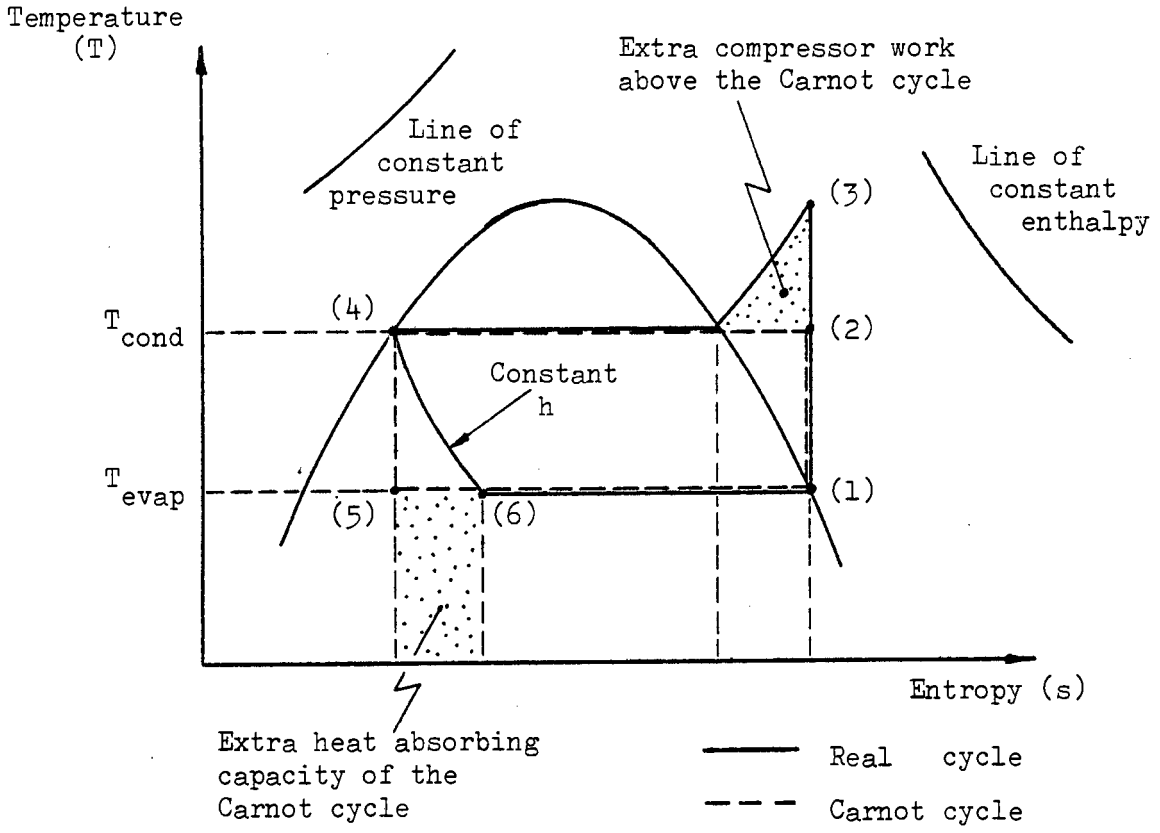
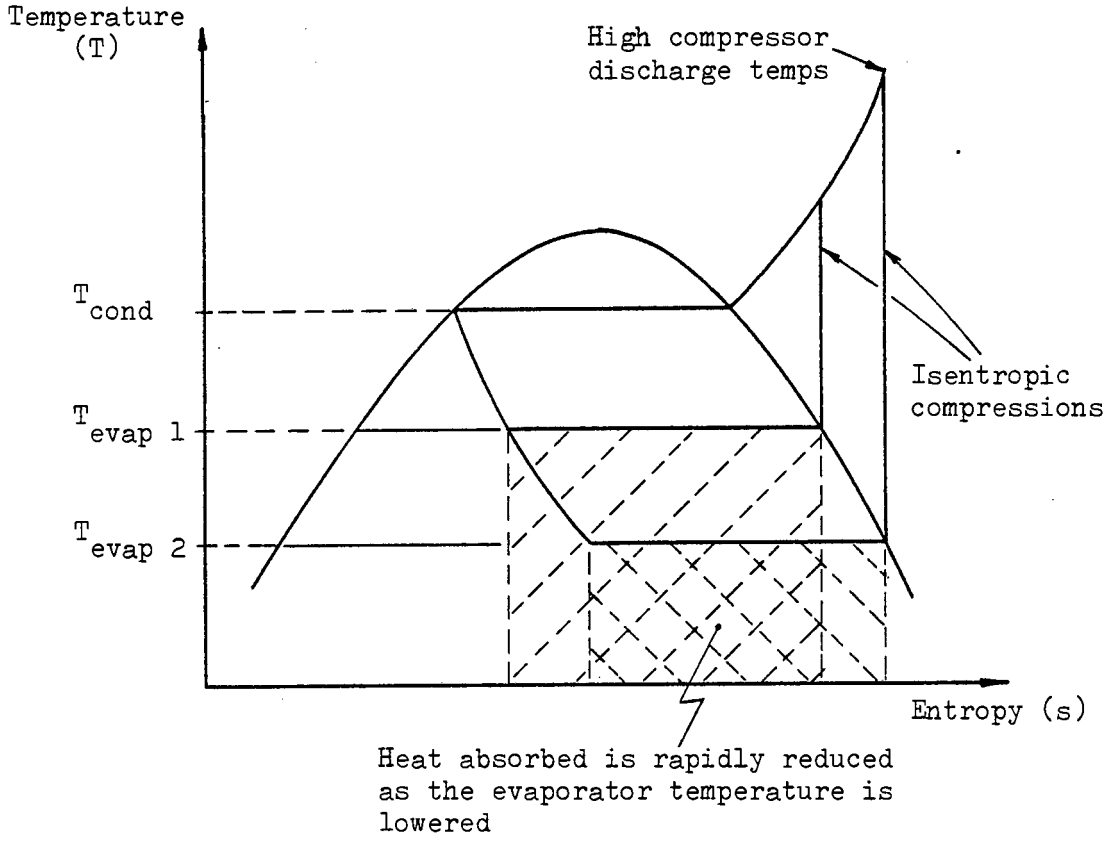


FIG A.6 THE EFFECT OF DECREASING THE EVAPORATOR TEMPERATURE OF A SINGLE STAGE V.C. CYCLE



A.4 THE TWO-STAGE V.C. CYCLE WITH INTERCOOLING

Equation A.3 shows that the coefficient of performance of the Carnot refrigeration cycle (and thus all other refrigeration cycles) decreases as :

i) The evaporating temperature decreases.

ii) The temperature difference between the evaporator and the condenser increases.

Thus for low evaporating temperatures poor coefficients of performance are obtained with single stage vapour compression refrigeration cycles. Cycles with two or three compression stages are therefore used in these cases and have the following advantages :

i) Increased relative performance i.e. the performance compared to a Carnot cycle between the end points is improved.

ii) Increased absolute performance i.e. the performance of the high and low pressure cycles is improved since the temperature differences for each are smaller.

iii) The final temperatures of compression are kept within reasonable limits.

Decreasing the evaporating temperature of a single stage cycle results in a rapid reduction in the heat absorbed by the refrigerant, very high temperatures at the end of compression and increased compressor work. These effects can be seen in Figure A.6. With the compressor work increased and the heat absorbed decreased the coefficient of performance is substantially reduced as a result of lowering the evaporator temperature.

The two-stage cycle is essentially two single stage cycles arranged so that the heat sink for the low pressure cycle is the heat source for the high pressure cycle. Figure A.7 shows a two stage V.C. cycle with intercooling. The system consists of a condenser, evaporator, flash tank, low pressure (l.p.) compressor and high pressure (h.p.) compressor. The flash tank operates at a pressure intermediate between the condenser and evaporator pressures. The l.p. compressor compresses the refrigerant vapour produced in the evaporator and the flash gas from the second expansion to the intermediate flash tank pressure. The discharge from this compressor is cooled by bubbling it through the liquid refrigerant in the flash tank some of which is evaporated to provide the cooling. The h.p. compressor takes vapour from the flash tank and compresses it to the condenser pressure. This vapour includes the l.p. compressor discharge, flash gas from the first expansion and the refrigerant evaporated in the flash tank to cool the l.p. compressor discharge. Figure A.8 shows a two stage V.C. cycle with intercooling on a temperature - entropy diagram. It can be seen from this diagram that the "superheat horns" are reduced and the high compressor discharge temperatures are eliminated as a result of introducing two stages. The refrigerating effect is also substantially increased by the introduction of a second stage. This increase can be clearly seen in Figure A.9 which shows the cycle on a pressure - enthalpy diagram.

FIG A.7 TWO STAGE VAPOUR COMPRESSION CYCLE WITH INTERCOOLING

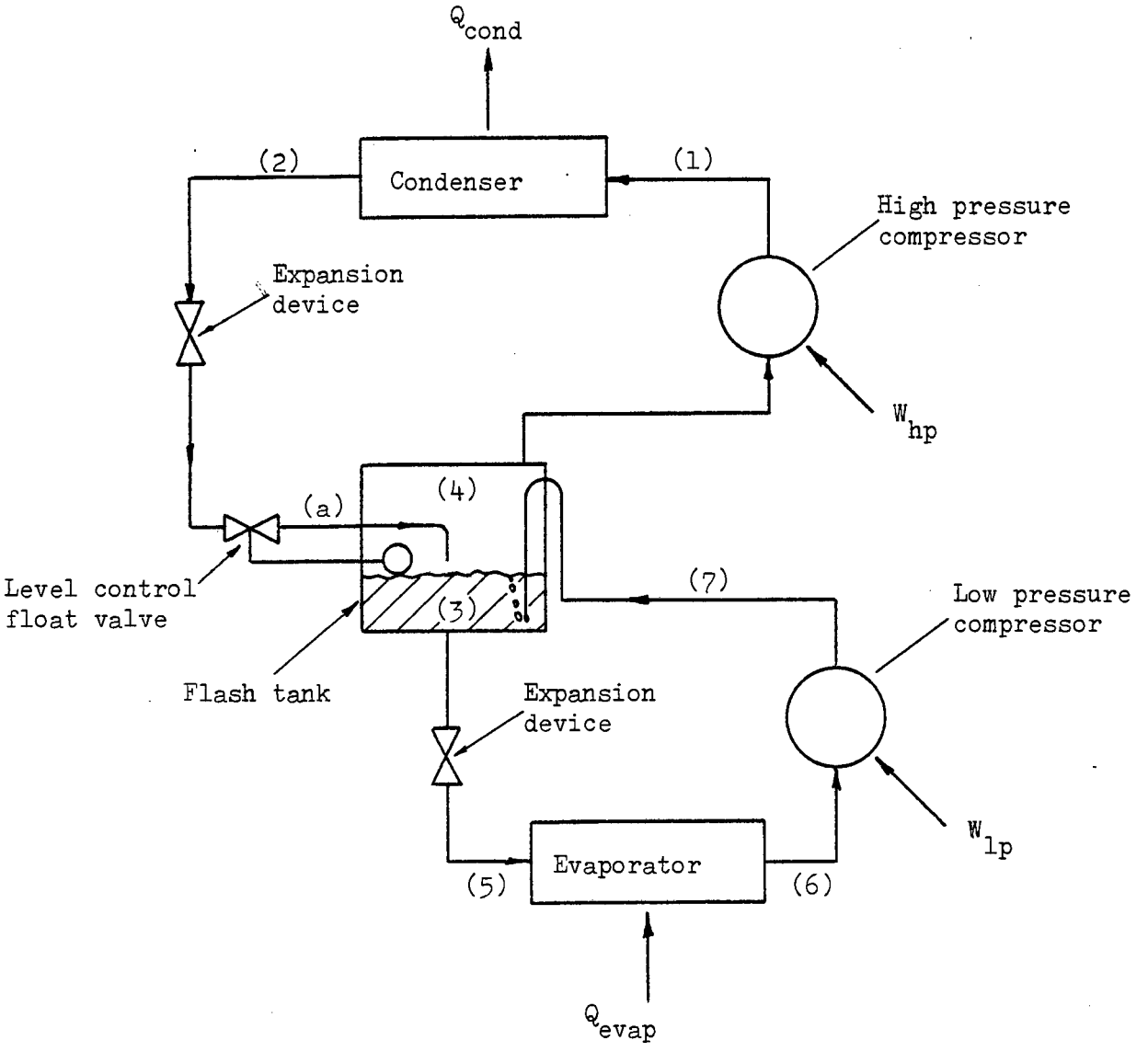


FIG A.8 TWO STAGE V.C. CYCLE WITH INTERCOOLING ON A TEMPERATURE - ENTROPY DIAGRAM

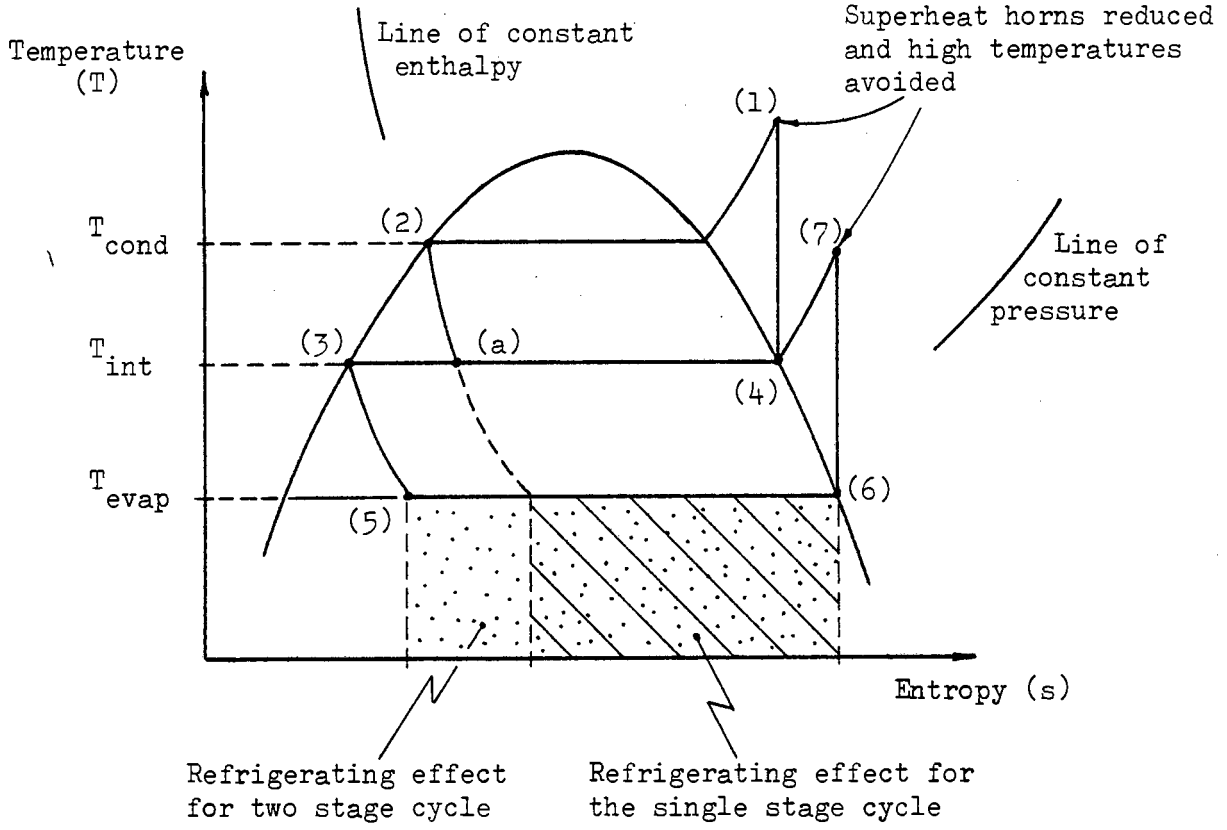
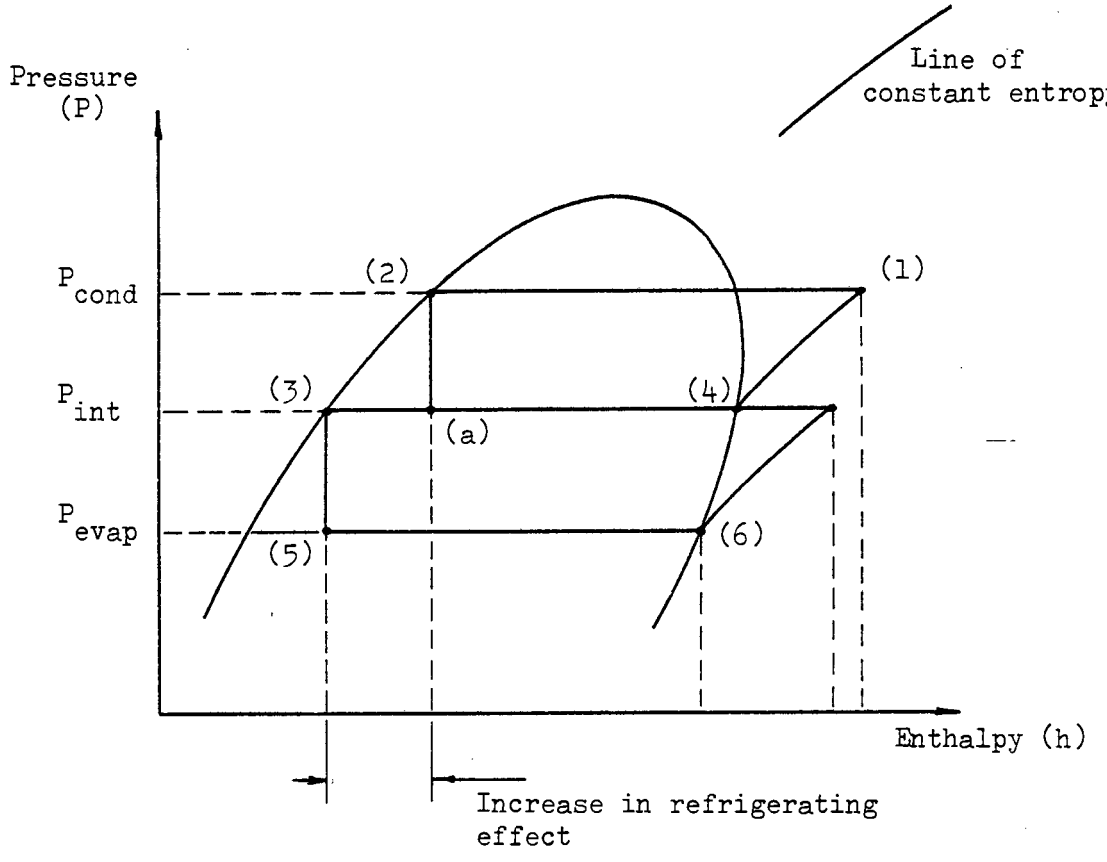


FIG A.9 TWO STAGE V.C. CYCLE WITH INTERCOOLING ON A PRESSURE - ENTHALPY DIAGRAM



The coefficient of performance of the two stage cycle with intercooling is given by the equation :

$$COP_{2\ stage} = Q_{evap}/(W_{lp} + W_{hp}) \quad --- \quad (A.6)$$

Where

$COP_{2\ stage}$ = Coefficient of performance of a two stage V.C. cycle with intercooling
W_{lp} = Low pressure compressor work (J/s).
W_{hp} = High pressure compressor work (J/s).
Q_{evap} = Heat absorbed in evaporator (J/s).

Refering to the cycle points labelled on Figures A.7 and A.9 the following equations can be developed for the two-stage cycle with intercooling :

$$W_{lp} = m_6 \cdot (h_6 - h_5) \quad --- \quad (A.7)$$
$$W_{hp} = m_1 \cdot (h_1 - h_4) \quad --- \quad (A.8)$$
$$m_f = m_1 \cdot (h_a - h_3)/(h_4 - h_3) \quad --- \quad (A.9)$$
$$m_c = m_6 \cdot (h_7 - h_4)/(h_4 - h_3) \quad --- \quad (A.10)$$
$$m_1 = m_6 + m_f + m_c \quad --- \quad (A.11)$$
$$Q_{evap} = m_6 \cdot (h_6 - h_5) \quad --- \quad (A.12)$$
$$Q_{cond} = m_1 \cdot (h_1 - h_2) \quad --- \quad (A.13)$$

Where

h_i = Enthalpy at cycle point i (J/kg).
m_i = Refrigerant mass flow rate at cycle point i (kg/s)
i = Cycle point number.
m_f = Mass flow rate of flash gas from the first expansion (kg/s).
m_c = Mass flow rate of refrigerant evaporated to cool l.p. compressor discharge (kg/s).
Q_{cond} = Heat rejected to the condenser (J/s).

Substituting these equations into equation (A.6) gives the equation :

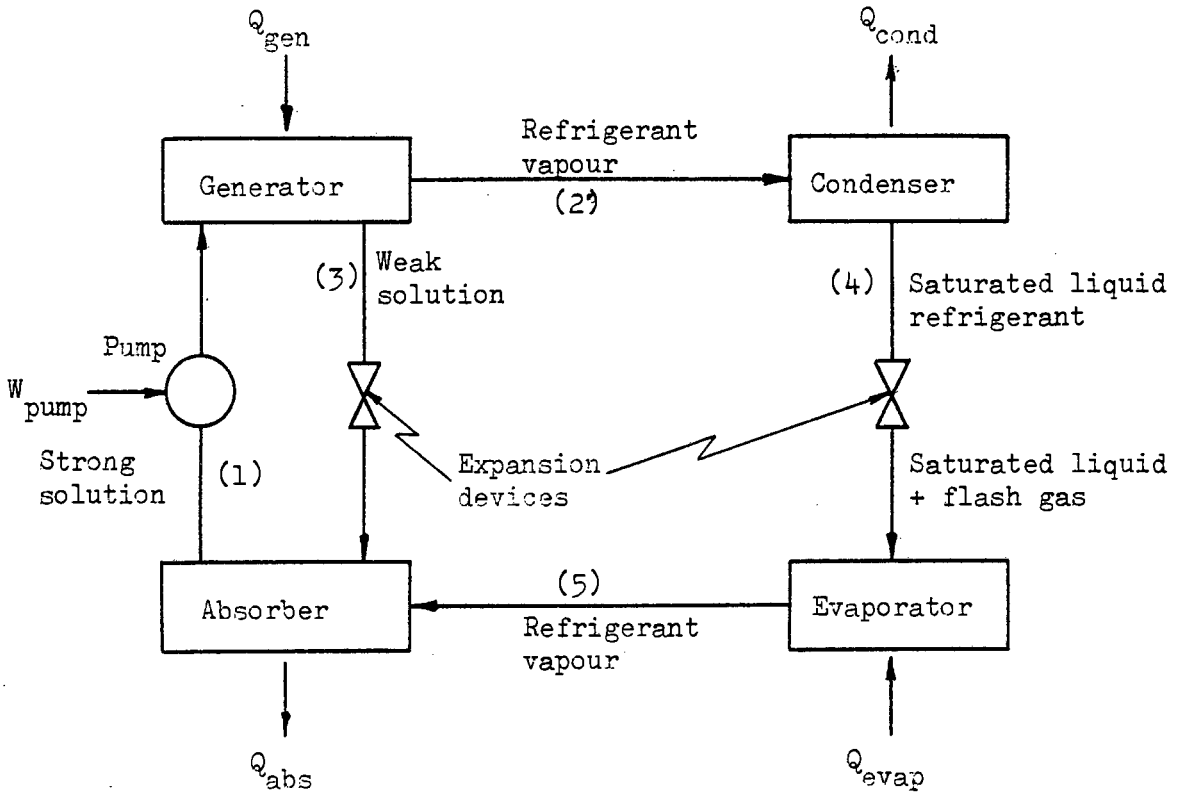
$$COP_{2\ stage} = \frac{m_6 \cdot (h_6 - h_5)}{m_1 \cdot (h_1 - h_4) + m_6 \cdot (h_7 - h_6)} \quad \text{---} \quad (A.14)$$

A.5 THE VAPOUR ABSORPTION REFRIGERATION CYCLE

The vapour absorption (V.A) refrigeration cycle differs from the vapour compression cycle in the method used to transport the refrigerant vapour from the evaporator to the condenser. Instead of a compressor the V.A. cycle uses a system comprising of a generator, a pump, an absorber and an expansion device.

Figure A.10 shows a simple V.A. refrigeration system [6]. Refering to this figure, saturated vapour from the evaporator is passed to the absorber where it is absorbed. The absorbent is a liquid with the property of being able to absorb vast quantities of the refrigerant. Absorbent from the absorber, which is rich in absorbed refrigerant, is pumped to the generator where it is heated. As the temperature of the absorbent is increased the solubility of the refrigerant in the absorbent decreases and some of the refrigerant is released. This refrigerant vapour is passed to the condenser where it is cooled and condensed. The condensed liquid refrigerant is expanded to the evaporating pressure and evaporated in the evaporator to produce the refrigerating effect, thereby completing the refrigerant path. The hot absorbent in the generator, which is now lean in absorbed refrigerant, is returned to the absorber where it is cooled and allowed to absorb more refrigerant vapour. It is then pumped back to the generator thus completing the absorbent path.

FIG A.10 SIMPLE ABSORPTION REFRIGERATION SYSTEM



The only mechanical work input required to this form of the V.A. cycle is that to the pump which, due to the low specific volume of the liquid absorbent, is usually negligibly small. However a considerable amount of low grade energy has to be supplied to the generator to heat the absorbent solution. This heat input is by far the largest energy input to the cycle and the coefficient of performance of the V.A. cycle is therefore defined as :

$$COP_{va} = Q_{evap}/Q_{gen} \qquad \text{---} \qquad (A.15)$$

Where

COP_{va} = Coefficient of performance of the vapour absorption cycle.

Q_{evap} = Heat absorbed in the evaporator (J/s).

Q_{gen} = Heat supplied to the generator (J/s).

In order to reduce the heat input to the generator and thereby increase the performance of the V.A. cycle, a regenerative heat exchanger is usually fitted to the system. Hot absorbent returning to the absorber is cooled in the heat exchanger and its heat is transferred to the cold absorbent being pumped to the generator. This results in less "new" energy having to be supplied to heat the generator and the coefficient of performance of the cycle is therefore increased.

A.5 MAXIMUM THEORECTICAL PERFORMANCE OF A VAPOUR ABSORPTION REFRIGERATION CYCLE

In order to establish the maximum theoretical performance of a V.A. cycle it is necessary to consider the hypothetical case of a Carnot heat engine, operating between the generator temperature and ambient temperature, driving a reversed Carnot engine (heat pump) operating between the evaporator temperature and ambient temperature. Figure A.11 shows this concept diagramatically.

From Carnot principles the efficiency of the heat engine and the performance of the reversed heat engine are given by the equations :

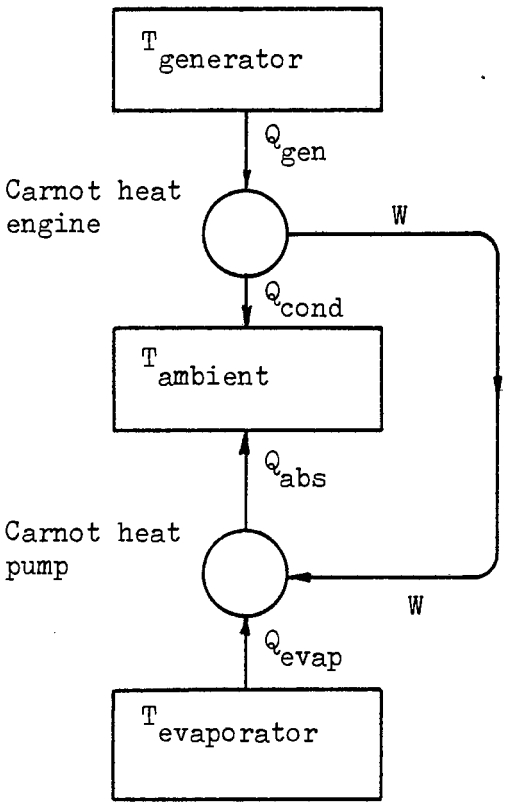
$$\eta_{HE} = W/Q_{gen} = (T_{gen} - T_{amb}) / T_{gen} \quad --- \quad (A.16)$$

$$\eta_{REF} = Q_{evap}/W = T_{evap}/(T_{amb} - T_{evap}) \quad --- \quad (A.17)$$

Where

η_{HE} = Efficiency of Carnot heat engine.
η_{REF} = Performance of Carnot heat pump.
W = Carnot engine work output (J/s).
= Work input to Carnot heat pump (J/s).
Q_{evap} = Heat absorbed in the evaporator (J/s).
Q_{gen} = Heat supplied to the generator (J/s).
T_{evap} = Evaporator temperature (K).
T_{cond} = Condenser temperature (K).
T_{amb} = Ambient temperature (K).

FIG A.11 DIAGRAMATIC REPRESENTATION OF A CARNOT HEAT ENGINE DRIVING A CARNOT HEAT PUMP



But from equation (A.15)

$$COP_{v.a.} = Q_{evap}/Q_{gen}$$

So from equations (A.16) and (A.17)

$$COP_{v.a.\ max} = \eta_{HE} \cdot \eta_{REF} \qquad \text{---} \quad (A.18)$$

And so

$$COP_{v.a.\ max} = \frac{T_{evap}\cdot(T_{gen}-T_{amb})}{T_{gen}\cdot(T_{amb}-T_{evap})} \qquad \text{---} \quad (A.19)$$

Where

$COP_{v.a.}$ = Coeff of performance of V.A. cycle.

$COP_{v.a.\ max}$ = Maximum possible coefficient of performance of V.A. cycle.

APPENDIX B

THEORETICAL EVALUATION OF A REFRIGERATION CYCLE WITH MULTIPLE EXPANSIONS AND COMPRESSIONS

B.1 THEORETICAL COEFFICIENT OF PERFORMANCE

In the multiple expansion and compression refrigeration cycle, as shown in Figure B.1 on a pressure enthalpy diagram, the saturated liquid from the condenser is expanded to the evaporator pressure in a series of small expansions. Each expansion results in the formation of flash gas (saturated vapour) and saturated liquid at a lower pressure. The flash gas is separated from the liquid and compressed to the condenser pressure directly. Less work is expended in compressing the flash gas from the stage pressures than if the refrigerant was fully expanded before the flash gas was compressed to the condenser pressure.

The coefficient of performance of the cycle can be defined as :

$$COP_{\text{multi exp}} = Q_{evap}/W_{tot} \qquad \text{---} \quad (B.1)$$

Where

$COP_{\text{multi exp}}$ = Coefficient of performance of a refrigeration cycle with multiple expansions and compressions

Q_{evap} = Heat into the evaporator (J/s)

W_{tot} = Total compressor work (J/s)

The total compressor work is given by the equation :

$$W_{tot} = W_{main} + \sum_{i=1}^{n} dW_i \qquad \text{---} \quad (B.2)$$

Where

W_{main} = Main compressor work i.e. work to compress the vapour from the evaporator (J/s)

dW_i = Compressor work to compress the flash gas from the i^{th} expansion

n = Number of expansion stages

Substituting equation (B.2) into (B.1) gives :

$$COP_{multi\ exp} = \frac{Qevap}{W_{main} + \sum_{i=1}^{n} dW_i} \qquad \text{--- (B.3)}$$

From equation (B.1) it can be seen that, as the total compressor work decreases, the coefficient of performance increases. As the number of expansion stages is increased, more flash gas is compressed through smaller pressure differentials. This results in a reduction in the compressor work and an increase in the coefficient of performance. The maximum possible performance would be obtained with an infinite number of expansion stages, each with infinitely small pressure drops.

The ideal compressor work required for each stage can be calculated from the equation :

$$dW_i = m_i \cdot dh_i \qquad \text{--- (B.4)}$$

Where

dW_i = Ideal compressor work for the i^{th} stage

m_i = Vapour mass flow rate at i^{th} stage

dh_i = Enthalpy rise during i^{th} compression

The main compressor work i.e. the compressor work to compress the refrigerant which is evaporated in the evaporator, can be calculated using the equation :

$$W_{main} = m_{evap} \cdot (h_2 - h_1) \qquad \text{---} \quad (B.5)$$

Where

W_{main} = Main compressor work (J/s)
m_{evap} = Evaporator mass flow rate (kg/s)
h_i = Enthalpy at cycle point i (see Fig B.1)

FIG B.1 MULTIPLE EXPANSION VAPOUR COMPRESSION REFRIGERATION CYCLE ON A PRESSURE - ENTHALPY DIAGRAM

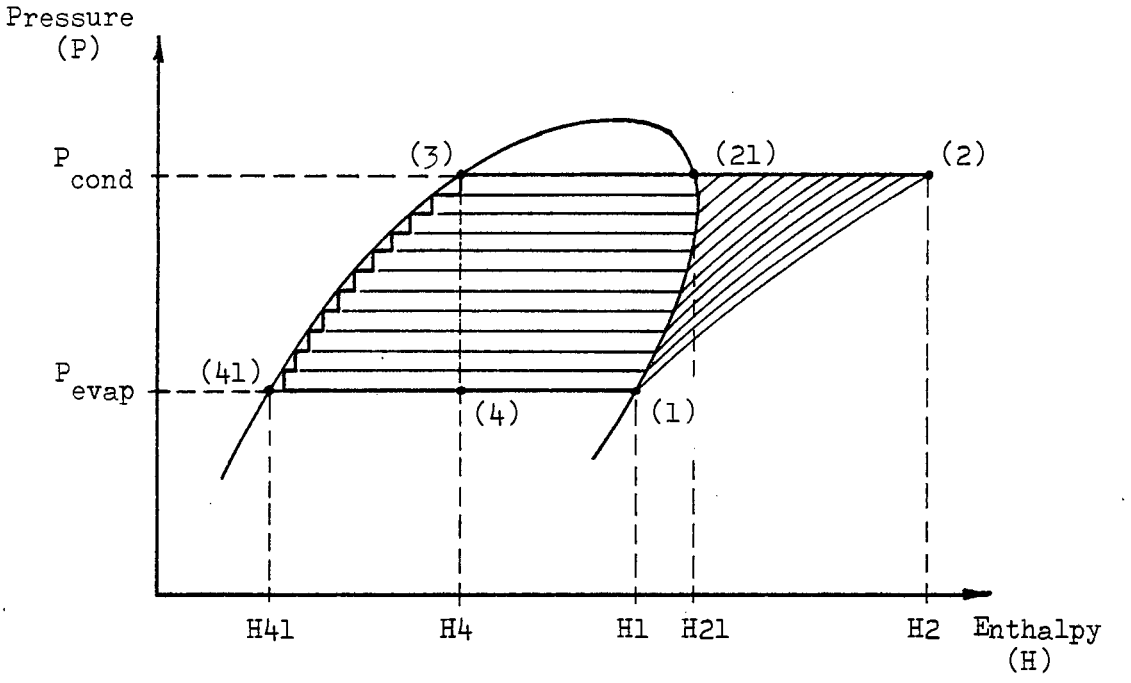
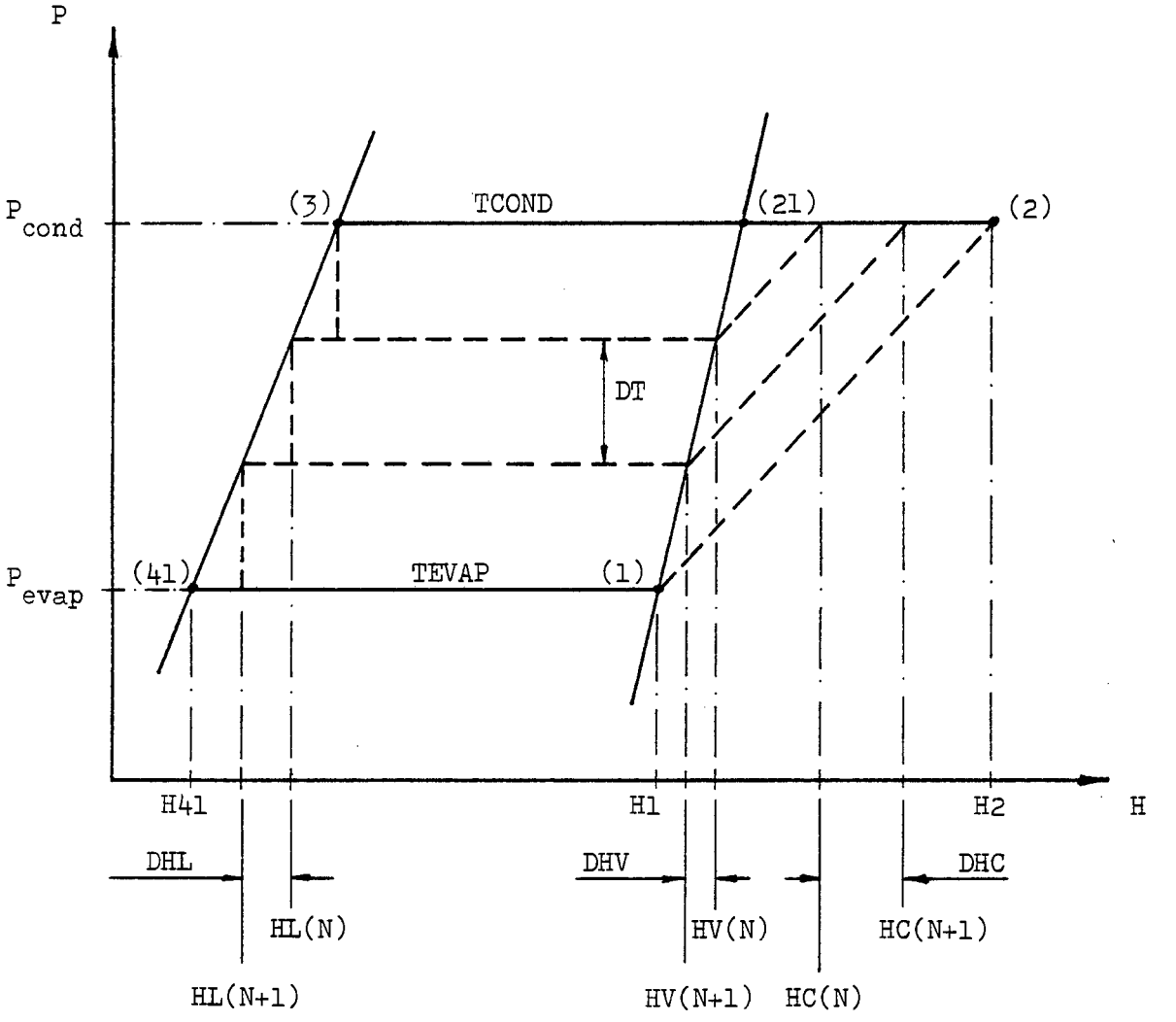


FIG B.2 MULTIPLE EXPANSION VAPOUR COMPRESSION REFRIGERATION CYCLE ON A SIMPLIFIED PRESSURE - ENTHALPY DIAGRAM



B.2 DETERMINATION OF THE COEFFICIENT OF PERFORMANCE

Due to the large number of stages and the repetitive nature of the calculations the determination of the theoretical coefficient of performance of the multiple expansion cycle, was suited to being performed by a micro-computer.

In order to calculate the ideal compressor work at each stage it was necessary to determine the following :

i) The refrigerant vapour fraction for each expansion stage and hence the vapour and liquid mass flow rates at each stage.

ii) The enthalpy of the saturated liquid and saturated vapour at each stage.

iii) The enthalpy of the flash gas from each expansion stage after an isentropic compression to the condenser pressure.

It was impractical to obtain all of the refrigerant enthalpies required, other than those at the main cycle points shown on Figure B.1, from the pressure - enthalpy diagram. It was therefore necessary to make some simplifying assumptions to enable the enthalpies required for the calculations to be more conveniently determined.

The following assumptions were therefore made :

i) The saturated liquid and vapour lines on the refrigerant pressure enthalpy diagram were assumed to be straight over the temperature range of interest.

ii) The lines of constant entropy on the pressure - enthalpy diagram were assumed to be straight and parallel in the region of interest.

With the above assumptions, the multiple expansion cycle of Figure B.1 became that shown in Figure B.2. It was then possible, given the enthalpies at the main cycle points, i.e. points 1, 2, 21, 3, and 41, to calculate any intermediate enthalpy required using linear interpolation.

Refering to Figure B.2 and the symbols on that diagram the following equations were formed :

(The equations have been presented as used in the computer programme and the syntax therefore corresponds to that of the BASIC computer programming language.)

On the saturated liquid line :

```
        DT/DHL  = (TCOND-TEVAP)/(H3-H41)

Thus    HL(N-1) = HL(N) + ( DT*(H3-H41)/(TCOND-TEVAP) )
```

On the saturated vapour line :

```
        DT/DHV  = (TCOND-TEVAP)/(H21-H1)

Thus    HV(N-1) = HV(N) + ( DT*(H21-H1)/(TCOND-TEVAP) )
```

For straight and parallel lines of constant entropy :

DHC = (H2-H21)/S

Thus HC(N-1) = HC(N) - DHC

Where

S = number of expansion stages

For each expansion stage :

X(N) = (HL(N-1)-HL(N))/(HV(N)-HL(N))

ML(N-1) = ML(N)/(1-X(N))

MV(N) = ML(N-1)*X(N)

W(N) = MV(N)*(HC(N)-HV(N))

Where

X(N) = Vapour fraction for the N^{th} stage
W(N) = Ideal compressor work for the N^{th} stage

For the evaporator :

QIN = ML(S+1)*(H1-H41)

ML(S+1) = QIN/(H1-H41)

WCOMP = ML(S+1)*(H2-H1)

Where

QIN = Heat into evaporator
ML(S+1) = Evaporator mass flow rate (initial known flow rate given QIN)
WCOMP = Main compressor work

For the cycle :

COP = QIN/WTOT

COP = QIN/(WCOMP+WTOT2)

Where

COP = Coefficient of performance
WTOT = Total compressor work
WTOT2 = Sum of all the stage compressor work

The above equations, enclosed in appropriate loops and supporting routines, were incorporated into the programme listed in section B.3, which calculated the coefficient of performance of the multi -expansion cycle. Details of the programme have been given in section B.3

B.3 PROGRAMME LISTING

The programme listed below was used to calculate the coefficient of performance of the multiple expansion vapour compression refrigeration cycle as described previously. Additional coding calculated, for comparison, the coefficients of performance of a Carnot cycle, a single stage V.C. cycle and a two stage V.C. cycle with intercooling. All of the cycles were considered between the same evaporating and condensing temperatures.

The programme required initialisation with the following data :

i) The evaporating pressure and temperature.

ii) The condensing pressure and temperature.

iii) The refrigerant used and the refrigerant enthalpies at the main cycle points shown in Figure B.2.

iv) The number of expansion stages to be considered

```
10 REM THIS PROGRAM CALCULATES THE COP OF A REFRIGERATION CYCLE
20 REM EMPLOYING MULTIPLE REFRIGERANT EXPANSIONS, WITH THE
30 REM FLASH GAS BEING COMPRESSED TO CONDENSER PRESSURE FROM THE
40 REM INTERMEDIATE PRESSURES AT WHICH IT IS FORMED
50 REM
60 REM DEVELOPED BY J A BROWNE, UCT, 28/3/85
70 REM REWRITTEN FOR IBM PC 4 OCTOBER 1985
80 '
90 REM DIMENSION ARRAYS
100 '
110 DIM HL(200)
120 DIM HV(200)
130 DIM T(200)
140 DIM W(200)
150 DIM X(200)
160 DIM MV(200)
170 DIM ML(200)
180 DIM HC(200)
190 DIM P(200)
200 '
210 REM INITIALISE ENTHALPIES (OBTAIN FROM RELEVANT P-h DIAG :UNITS
KJ/KG)
220 H1=246.6
230 H2=279.1
240 H21=260.1
250 H3=81.4
260 H4=H3
270 H41=32.6
```

```
280 '
290 REM RECORD DEFAULT REFRIGERANT TYPE & NO. OF EXPANSION STAGES
300 REF$="FREON 22" :S=10
310 '
320 REM INITIALISE DEFAULT TEMP LIMITS CHOSEN (UNITS - DEG KELVIN )
330 TCOND=305
340 TEVAP=265
350 '
360 REM INITIALISE DEFAULT CONDENSER & EVAPORATOR PRESSURES (UNITS -
BAR)
370 PCOND = 13.1
380 PEVAP = 6.1
390 '
400 REM INITIALISE HEAT INPUT CHOSEN (UNITS ARE KW )
410 QIN=1!
420 '
430 CLS
440 PRINT "START PROGRAMME EXECUTION"
450 PRINT
460 PRINT "SELECT OUTPUT DEVICE REQD - SCREEN [S],PRINTER [P],DISC
[D] "
470 INPUT A$
480 IF A$="P" THEN OPEN "LPT1:" FOR OUTPUT AS #1
490 IF A$="D" THEN OPEN "DATA1" FOR APPEND AS #1
500 IF A$="S" THEN OPEN "SCRN:" FOR OUTPUT AS #1
510 PRINT
520 PRINT "DETAILED OUTPUT REQD ?? (Y/N) "
530 INPUT B$
540 PRINT
550 PRINT "CHANGE PARAMETERS ?? (Y/N)"
560 INPUT C$
570 IF C$="Y" THEN PRINT ELSE GOTO 770
580 PRINT "CHANGE NO. OF EXPANSION STAGES ONLY  ?? (Y/N)"
590 INPUT E$
600 IF E$="Y" THEN GOTO 720 ELSE PRINT
610 PRINT "ENTER TEMPERATURES: [TCOND,TEVAP]    UNITS: KELVIN"
620 INPUT TCOND,TEVAP
630 PRINT
640 PRINT "ENTER PRESSURES: [PCOND,PEVAP]    UNITS: BAR"
650 INPUT PCOND,PEVAP
660 PRINT
670 PRINT "ENTER ENTHALPIES : [H1,H2,H21,H3,H4,H41]    UNITS: KJ/KG"
680 INPUT H1,H2,H21,H3,H4,H41
690 PRINT
700 PRINT "ENTER QIN AND REFRIGERANT TYPE: [QIN,REF]   UNITS FOR
QIN: KW"
710 INPUT QIN,REF$
720 PRINT
730 PRINT "ENTER NO. OF EXP STAGES: [S]"
740 INPUT S
750 IF S > 199 THEN BEEP : PRINT "ARRAYS NOT BIG ENOUGH" : GOTO 730
760 '
770 REM  INITIALISE OTHER VARIABLES
780 DT = (TCOND-TEVAP)/S
790 T(1)=TCOND
```

```
800 T(S+1)=TEVAP
810 HL(1)=H3
820 HV(1)=H21
830 HL(S+1)=H41
840 HV(S+1)=H1
850 HC(S+1)=H2
860 X(1)=0
870 W(1)=0
880 HC(1)=H21
890 MVTOT=0
900 WTOT2=0
910 P(S+1)=PEVAP
920 P(1)=PCOND
930 '
940 REM PRINT INITIAL DATA
950 PRINT #1,
960 PRINT #1,USING "REFRIGERANT = &         QIN = ##.###
KJ/S";REF$,QIN
970 PRINT #1,
980 PRINT #1,"ENTHALPIES (KJ/KG) "
990 PRINT #1,
1000 PRINT #1,USING "H1  = ###.#";H1
1010 PRINT #1,USING "H2  = ###.# ";H2
1020 PRINT #1,USING "H3  = ###.# ";H3
1030 PRINT #1,USING "H4  = ###.# ";H4
1040 PRINT #1,USING "H41 = ###.# ";H41
1050 PRINT #1,USING "H21 = ###.# ";H21
1060 PRINT #1,
1070 PRINT #1,"TEMPERATURES (KELVIN)          PRESSURES (BAR)"
1080 PRINT #1,
1090 PRINT #1,USING"TCOND = ###.#                  PCOND =
##.#";TCOND,PCOND
1100 PRINT #1,USING"TEVAP = ###.#                  PEVAP =
##.#";TEVAP,PEVAP
1110 PRINT #1,USING"DEL T = ###.# ";DT
1120 PRINT #1,
1130 PRINT #1,USING "NO. OF EXPANSION STAGES = ###";S
1140 PRINT #1,
1150 PRINT #1,
1160 '
1170 PRINT
1180 PRINT "WAIT ......."
1190 '
1200 REM CALC COP OF MULTIPLE EXPANSION CYCLE
1210 DHC=(H2-H21)/S
1220 ML(S+1)=QIN/(H1-H41)
1230 R1=(H3-H41)/(TCOND-TEVAP)
1240 R2=(H21-H1)/(TCOND-TEVAP)
1250   FOR N=S+1 TO 2 STEP -1
1260   IF N=2 THEN GOTO 1330
1270   HL(N-1)=HL(N)+(R1*DT)
1280   HV(N-1)=HV(N)+(R2*DT)
1290   T(N-1)=T(N)+DT
1300   R=( T(N-1)-TEVAP )/( TCOND-TEVAP )
1310   LP=LOG(PEVAP)+(LOG(PCOND)-LOG(PEVAP))*R
```

```
1320   P(N-1)=EXP(LP)
1330   X(N)=( HL(N-1)-HL(N) )/( HV(N)-HL(N) )
1340   ML(N-1)=ML(N)/(1-X(N))
1350   MV(N)=ML(N-1)*X(N)
1360   IF N=2 THEN GOTO 1380
1370   HC(N-1)=HC(N)-DHC
1380   W(N)=MV(N)*(HC(N)-HV(N))
1390   WTOT2=WTOT2+W(N)
1400   MVTOT=MVTOT+MV(N)
1410   NEXT N
1420 WCOMP=ML(S+1)*(H2-H1)
1430 WTOT=WTOT2+WCOMP
1440 W1=WCOMP/WTOT*100
1450 W2=WTOT2/WTOT*100
1460 COPIM=QIN/WTOT
1470 '
1480 REM CALC COP OF CARNOT CYCLE
1490 COPCAR=TEVAP/(TCOND-TEVAP)
1500 '
1510 REM CALC COP OF SINGLE STAGE CYCLE
1520 MTOT=QIN/(H1-H4)
1530 COPSSVC=QIN/(MTOT*(H2-H1))
1540 '
1550 REM CALC COP OF 2 STAGE CYCLE WITH INTERCOOLING
1560 DT2=(TCOND-TEVAP)/2
1570 H8=H21-(R2*DT2)
1580 H5=H3-(R1*DT2)
1590 H6=H5
1600 H9=(H2-H21)/2+H21
1610 H7=(H2-H1)/2+H1
1620 ME=QIN/(H1-H6)
1630 MCOOL=ME*(H7-H8)/(H8-H5)
1640 MC=(ME+MCOOL)/(1-(H3-H5)/(H8-H5))
1650 MF=MC*(H3-H5)/(H8-H5)
1660 WLP=ME*(H7-H1)
1670 WHP=MC*(H9-H8)
1680 COP2SIC=QIN/(WLP+WHP)
1690 '
1700 REM CALC % IMPROVEMENT IN COPMEXP COMPARED TO OTHER CYCLE COP'S
1710 IMOVERSS=100*(COPIM-COPSSVC)/COPSSVC
1720 IMOVER2S=100*(COPIM-COP2SIC)/COP2SIC
1730 '
1740 REM PRINT DATA TABLE IF REQD
1750 IF B$="N" OR B$="NO" THEN GOTO 1870
1760 PRINT #1,
1770 PRINT #1,"TABLE OF TEMPS,ENTHALPIES & MASS FLOW RATES AT EACH
EXPANSION STAGE"
1780 PRINT #1,
1790 PRINT #1,          "   N   TEMP    PRESS     HL        HV
HC         X        MV        W"
1800 PRINT #1,          "       (K)     (BAR)    (KJ/KG)   (KJ/KG)
(KJ/KG)              (KG/S)   (J/S)"
1810 PRINT #1,
1820 PRINT #1,
1830   FOR M=1 TO S+1 STEP 1
```

```
1840   PRINT #1 ,USING " ###   ###.##  ##.##   ###.##   ###.##
###.##   ##.##^^^^ ##.##^^^^
##.###";M,T(M),P(M),HL(M),HV(M),HC(M),X(M),MV(M),W(M)*1000
1850 PRINT #1,
1860   NEXT M
1870 '
1880 REM PRINT REFRIGERANT MASS FLOW RATES
1890 PRINT #1,      "REFRIGERANT MASS FLOW RATES (KG/S)"
1900 PRINT #1,
1910 PRINT #1,USING "M EVAP      = ##.###^^^^  " ; ML(S+1)
1920 PRINT #1,USING "M COND      = ##.###^^^^  " ; MTOT
1930 PRINT #1,USING "M FLASH VAP = ##.###^^^^  " ; MVTOT
1940 PRINT #1,
1950 PRINT #1,"COMPRESSOR WORK REQUIRED (J/S)
1960 PRINT #1,
1970 REM PRINT THE VARIOUS COMPRESSOR WORK REQD
1980 PRINT #1,USING "MAIN           = ####.##   % TOTAL =
##.##";WCOMP*1000,W1
1990 PRINT #1,USING "SEC(FLASH VAP) = ####.##   % TOTAL =
##.##";WTOT2*1000,W2
2000 PRINT #1,USING "TOTAL          = ####.##";WTOT*1000
2010 '
2020 REM PRINT COP'S
2030 PRINT #1,
2040 PRINT #1,"COEFFICIENTS OF PERFORMANCE"
2050 PRINT #1,
2060 PRINT #1,USING"COPCARNOT = ##.##";COPCAR
2070 PRINT #1,USING"COPMULEXP = ##.##";COPIM
2080 PRINT #1,USING"COP2SIC   = ##.##";COP2SIC
2090 PRINT #1,USING"COPSSVC   = ##.##";COPSSVC
2100 PRINT #1,
2110 PRINT #1,USING"IMPROVEMENT IN COP OVER SINGLE STAGE     = ##.#
%";IMOVERSS
2120 PRINT #1,USING"IMPROVEMENT IN COP OVER 2 STAGE INTCOOL  = ##.#
%";IMOVER2S
2130 PRINT #1,
2140 PRINT #1,
2150 CLOSE #1
2160 PRINT
2170 PRINT "RE-RUN PROGRAMME ?? (Y/N)"
2180 INPUT D$
2190 IF D$="Y" THEN CLS :GOTO 450
2200 PRINT
2210 PRINT "END PROGRAMME EXECUTION"
2220 END
```

B.4 TYPICAL RESULTS PRODUCED BY THE PROGRAMME

PRINTOUT 1:

REFRIGERANT = REF 22 QIN = 0.860 KJ/S

ENTHALPIES (KJ/KG)

H1 = 246.6
H2 = 281.5
H3 = 86.1
H4 = 86.1
H41 = 32.6
H21 = 260.5

TEMPERATURES (KELVIN)	PRESSURES (BAR)
TCOND = 308.0	PCOND = 13.1
TEVAP = 263.0	PEVAP = 3.5
DEL T = 9.0	

NO. OF EXPANSION STAGES = 5

TABLE OF TEMPS,ENTHALPIES & MASS FLOW RATES AT EACH EXPANSION STAGE

N	TEMP (K)	PRESS (BAR)	HL (KJ/KG)	HV (KJ/KG)	HC (KJ/KG)	X	MV (KG/S)	W (J/S)
1	308.00	13.10	86.10	260.50	260.50	0.00E+00	0.00E+00	0.000
2	299.00	10.03	75.40	257.72	264.70	5.87E-02	3.12E-04	2.175
3	290.00	7.68	64.70	254.94	268.90	5.62E-02	2.81E-04	3.924
4	281.00	5.88	54.00	252.16	273.10	5.40E-02	2.55E-04	5.333
5	272.00	4.51	43.30	249.38	277.30	5.19E-02	2.32E-04	6.468
6	263.00	3.45	32.60	246.60	281.50	5.00E-02	2.12E-04	7.382

REFRIGERANT MASS FLOW RATES (KG/S)

M EVAP = 4.019E-03
M COND = 5.358E-03
M FLASH VAP = 1.291E-03

COMPRESSOR WORK REQUIRED (J/S)

MAIN = 140.25 % TOTAL = 84.73
SEC(FLASH VAP) = 25.28 % TOTAL = 15.27
TOTAL = 165.53

COEFFICIENTS OF PERFORMANCE

COPCARNOT = 5.84
COPMULEXP = 5.20
COP2SIC = 4.83
COPSSVC = 4.60

IMPROVEMENT IN COP OVER SINGLE STAGE = 13.0 %
IMPROVEMENT IN COP OVER 2 STAGE INTCOOL = 7.6 %

PRINTOUT 2:

```
REFRIGERANT = REF 22          QIN =  0.860  KJ/S

ENTHALPIES (KJ/KG)

H1  = 246.6
H2  = 281.5
H3  =  86.1
H4  =  86.1
H41 =  32.6
H21 = 260.5

TEMPERATURES (KELVIN)              PRESSURES (BAR)

TCOND = 308.0                      PCOND = 13.1
TEVAP = 263.0                      PEVAP =  3.5
DEL T =   0.9

NO. OF EXPANSION STAGES =  50


REFRIGERANT MASS FLOW RATES (KG/S)

M EVAP      =  4.019E-03
M COND      =  5.358E-03
M FLASH VAP =  1.281E-03

COMPRESSOR WORK REQUIRED (J/S)

MAIN           =  140.25   % TOTAL = 86.97
SEC(FLASH VAP) =   21.01   % TOTAL = 13.03
TOTAL          =  161.26

COEFFICIENTS OF PERFORMANCE

COPCARNOT =  5.84
COPMULEXP =  5.33
COP2SIC   =  4.83
COPSSVC   =  4.60

IMPROVEMENT IN COP OVER SINGLE STAGE     = 16.0 %
IMPROVEMENT IN COP OVER 2 STAGE INTCOOL  = 10.5 %
```

PRINTOUT 3:

REFRIGERANT = REF 22 QIN = 0.860 KJ/S

ENTHALPIES (KJ/KG)

H1 = 246.6
H2 = 281.5
H3 = 86.1
H4 = 86.1
H41 = 32.6
H21 = 260.5

TEMPERATURES (KELVIN) PRESSURES (BAR)

TCOND = 308.0 PCOND = 13.1
TEVAP = 263.0 PEVAP = 3.5
DEL T = 0.5

NO. OF EXPANSION STAGES = 100

REFRIGERANT MASS FLOW RATES (KG/S)

M EVAP = 4.019E-03
M COND = 5.358E-03
M FLASH VAP = 1.280E-03

COMPRESSOR WORK REQUIRED (J/S)

MAIN = 140.25 % TOTAL = 87.10
SEC(FLASH VAP) = 20.77 % TOTAL = 12.90
TOTAL = 161.03

COEFFICIENTS OF PERFORMANCE

COPCARNOT = 5.84
COPMULEXP = 5.34
COP2SIC = 4.83
COPSSVC = 4.60

IMPROVEMENT IN COP OVER SINGLE STAGE = 16.1 %
IMPROVEMENT IN COP OVER 2 STAGE INTCOOL = 10.6 %

APPENDIX C

** EXPERIMENTAL DATA SHEET **

MULTIPLE EXPANSION - ABSORPTION MODE

SHEET No. : 2A

DATE : 26/8/86

Atm Press : 752,2 mm Hg
0,999 bar

REMARKS : .....
Mass bucket = 0,45 kg
Thermostat set at 100 deg C
Comp inlet pressure = 18 psi
.....
.....

PRESSURES :

LOCATION	PRESSURE (bar)		TEMP
	Gauge	Abs	(Deg Cel)
Condenser	8,50	9,50	21,32
Stage 1	7,40	8,40	16,91
Stage 2	6,40	7,40	12,51
Stage 3	4,50	5,50	2,66
Stage 4	2,70	3,70	- 9,99
Stage 5	1,50	2,50	-19,73
Generator	8,50	9,50	---
Absorber	1,10	2,10	---

TEMPERATURES:

LOCATION	mV	Deg C
1	0,60	14,81
2	0,52	12,84
3	-0,62	-15,31
4	-0,40	- 9,88
5	0,60	14,81
6	0,48	11,85
7	0,09	- 7,65

LOCATION	mV	Deg C
8	-0,31	- 7,65
9	-0,72	-17,78
10	4,03	99,51
11	0,63	15,56
12	0,61	15,06
13	0,74	18,27
14	0,63	15,56

COOLING & HEATING WATER FLOW RATES :

	MASS BUCKET + WATER (Kg)	TIME (mins)	FLOW RATE (Kg/s)
Condenser	6,24	9,50	$10,158 \times 10^{-3}$
Evaporator	5,64	3,00	$28,833 \times 10^{-3}$
Absorber	3,96	3,00	$19,500 \times 10^{-3}$

POWER INPUTS :

	MASS (Kg)	DIST (mm)	SPEED (rpm)	ELEC POWER (kWh)		TIME	
				Start	End	Start	End
Compressor	0,799	161	1470	8518,040	8518,342	15h30	16h30
Pump	---	---	---	1291,965	1291,977	15h30	16h30
Heater	---	---	---	3817,347	3817,424	15h300	16h00

** EXPERIMENTAL DATA SHEET **

SINGLE EXPANSION AND COMPRESSION MODE

SHEET No. : 2B
DATE : 26/8/86
Atm Press : 752,2 mm Hg
0,999 bar

REMARKS : Mass bucket = 0,45 kg
Comp inlet press = 18 psi

PRESSURES :

LOCATION	PRESSURE (bar)		TEMP (Deg Cel)
	Gauge	Abs	
Condenser	8,50	9,50	21,32

TEMPERATURES:

LOCATION	mV	Deg C
1	0,59	14,57
2	0,51	12,59
3	-0,62	-15,31
4	-0,41	-10,12
12	0,61	15,06
13	0,77	19,01
14	0,66	16,30

COOLING & HEATING WATER FLOW RATES :

	MASS BUCKET + WATER (Kg)	TIME (mins)	FLOW RATE (Kg/s)
Condenser	11,10	12,00	$14,792 \times 10^{-3}$
Evaporator	5,54	3,00	$28,278 \times 10^{-3}$

POWER INPUTS :

	MASS : DIST	SPEED	ELEC POWER (kWh)		TIME	
	(Kg) : (mm)	(rpm)	Start	End	Start	End
Compressor	0,799 : 158	1465	8518,455	8518,610	16h50	17h20

** EXPERIMENTAL DATA SHEET **

MULTIPLE EXPANSION - ABSORPTION MODE

SHEET No. : 3A
DATE : 27/8/86
Atm Press : 753,6 mm Hg
1,001 bar

REMARKS :
Mass bucket = 0,45 kg
Thermostat set at 120 deg C
Comp inlet press = 19 psi

PRESSURES :

LOCATION	PRESSURE (bar)		TEMP
	Gauge	Abs	(Deg Cel)
Condenser	8,30	9,30	20,39
Stage 1	7,30	8,30	16,70
Stage 2	5,90	6,90	10,22
Stage 3	4,30	5,30	1,53
Stage 4	2,50	3,50	-10,73
Stage 5	1,50	2,50	-19,73
Generator	8,30	9,30	---
Absorber	1,10	2,10	---

TEMPERATURES:

LOCATION	mV	Deg C
1	0,56	13,83
2	0,47	11,60
3	-0,76	-18,77
4	-0,55	-13,58
5	0,59	14,57
6	0,38	9,38
7	0,00	0,00

LOCATION	mV	Deg C
8	-0,44	-10,86
9	-0,74	-18,27
10	4,74	117,04
11	0,59	14,57
12	0,57	14,07
13	0,70	17,28
14	0,65	16,05

COOLING & HEATING WATER FLOW RATES :

	MASS BUCKET + WATER (Kg)	TIME (mins)	FLOW RATE (Kg/s)
Condenser	4,54	4,00	$16,208 \times 10^{-3}$
Evaporator	5,36	3,00	$27,278 \times 10^{-3}$
Absorber	5,96	5,00	$18,367 \times 10^{-3}$

POWER INPUTS :

	MASS (Kg)	DIST (mm)	SPEED (rpm)	ELEC POWER (kWh)		TIME	
				Start	End	Start	End
Compressor	0,799	159	1465	8519,171	8519,490	11h15	12h15
Pump	---	---	---	1291,991	1292,003	11h15	12h15
Heater	---	---	---	3817,779	3817,864	11h15	12h15

** EXPERIMENTAL DATA SHEET **

SINGLE EXPANSION AND COMPRESSION MODE

SHEET No. : 3B
DATE : 27/8/86
Atm Press : 753,6 mm Hg
1,001 bar

REMARKS : Mass bucket = 0,45 kg
Comp inlet press = 19 psi

PRESSURES :

LOCATION	PRESSURE (bar)		TEMP (Deg Cel)
	Gauge	Abs	
Condenser	8,40	9,40	21,11

TEMPERATURES:

LOCATION	mV	Deg C
1	0,55	13,58
2	0,46	11,36
3	-0,76	-18,77
4	-0,57	-14,07
12	0,57	14,07
13	0,71	17,53
14	0,64	15,80

COOLING & HEATING WATER FLOW RATES :

	MASS BUCKET + WATER (Kg)	TIME (mins)	FLOW RATE (Kg/s)
Condenser	5,25	5,00	$16,000 \times 10^{-3}$
Evaporator	4,65	3,00	$23,333 \times 10^{-3}$

POWER INPUTS :

	MASS : DIST	SPEED	ELEC POWER (kWh)		TIME	
	(Kg) : (mm)	(rpm)	Start	End	Start	End
Compressor	0,799 161	1471	8520,124	8520,362	14h15	15h00

**** EXPERIMENTAL DATA SHEET ****

MULTIPLE EXPANSION - ABSORPTION MODE

PRESSURES :

SHEET No. : 4A
DATE : 28/8/86
Atm Press : 754,2 mm Hg
1,002 bar

REMARKS : .....
Mass bucket = 0,45 kg
Comp inlet press = 19 psi
Thermostat set at 120 deg C

LOCATION	PRESSURE (bar)		TEMP
	Gauge	Abs	(Deg Cel)
Condenser	8,50	9,50	21,32
Stage 1	7,50	8,50	17,12
Stage 2	5,70	6,70	9,18
Stage 3	4,40	5,40	2,37
Stage 4	3,00	4,00	- 7,71
Stage 5	1,50	2,50	-19,73
Generator	8,50	9,50	---
Absorber	0,60	1,60	---

TEMPERATURES:

LOCATION	mV	Deg C
1	0,55	13,58
2	0,44	10,86
3	-0,79	-19,51
4	-0,56	-13,83
5	0,63	15,56
6	0,37	9,14
7	0,04	0,99

LOCATION	mV	Deg C
8	-0,32	- 7,90
9	-0,74	-18,27
10	4,77	117,78
11	0,59	14,57
12	0,59	14,57
13	0,73	18,02
14	0,68	16,79

COOLING & HEATING WATER FLOW RATES :

	MASS BUCKET + WATER (Kg)	TIME (mins)	FLOW RATE (Kg/s)
Condenser	6,64	7,00	$14,738 \times 10^{-3}$
Evaporator	7,03	6,00	$18,278 \times 10^{-3}$
Absorber	2,86	2,00	$20,083 \times 10^{-3}$

POWER INPUTS :

	MASS (Kg)	DIST (mm)	SPEED (rpm)	ELEC POWER (kWh)		TIME	
				Start	End	Start	End
Compressor	0,799	157	1470	8520,603	8520,920	10h00	11h00
Pump	---	---	---	1292,019	1292,027	10h00	11h00
Heater	---	---	---	3818,285	3818,373	10h00	11h00

**** EXPERIMENTAL DATA SHEET ****

SINGLE EXPANSION AND COMPRESSION MODE

PRESSURES :

LOCATION	PRESSURE (bar)		TEMP
	Gauge	Abs	(Deg Cel)
Condenser	8,50	9,50	21,32

SHEET No. : 4B
DATE : 28/8/86
Atm Press : 754,2 mm Hg
1,002 bar

REMARKS :
Mass bucket = 0,45 kg
Comp inlet press = 19psi

TEMPERATURES:

LOCATION	mV	Deg C
1	0,55	13,58
2	0,42	10,37
3	-0,80	-19,75
4	-0,58	-14,32
12	0,58	14,32
13	0,73	18,03
14	0,70	17,28

COOLING & HEATING WATER FLOW RATES :

	MASS BUCKET + WATER (Kg)	TIME (mins)	FLOW RATE (Kg/s)
Condenser	3,35	4,00	$12,083 \times 10^{-3}$
Evaporator	3,91	4,00	$14,417 \times 10^{-3}$

POWER INPUTS :

	MASS : DIST	SPEED	ELEC POWER (kWh)		TIME	
	(Kg) : (mm)	(rpm)	Start	End	Start	End
Compressor	0,799 : 157	1470	8521,231	8521,305	12h00	12h15

** THERMOCOUPLE LOCATIONS **

THERMOCOUPLE No.	LOCATION
1	Evaporator heating water inlet
2	Evaporator heating water outlet
3	Evaporator temperature
4	Evaporator vapour outlet
5	Stage 1 of expansion column (top)
6	Stage 2
7	Stage 3
8	Stage 4
9	Stage 5 of expansion column (bottom)
10	Generator
11	Absorber
12	Condenser cooling water inlet
13	Condenser cooling water outlet
14	Ambient temperature

International Standard Thermocouple Tables for

Nickel-chromium/Nickel-aluminium

Reference junction at 0 deg C

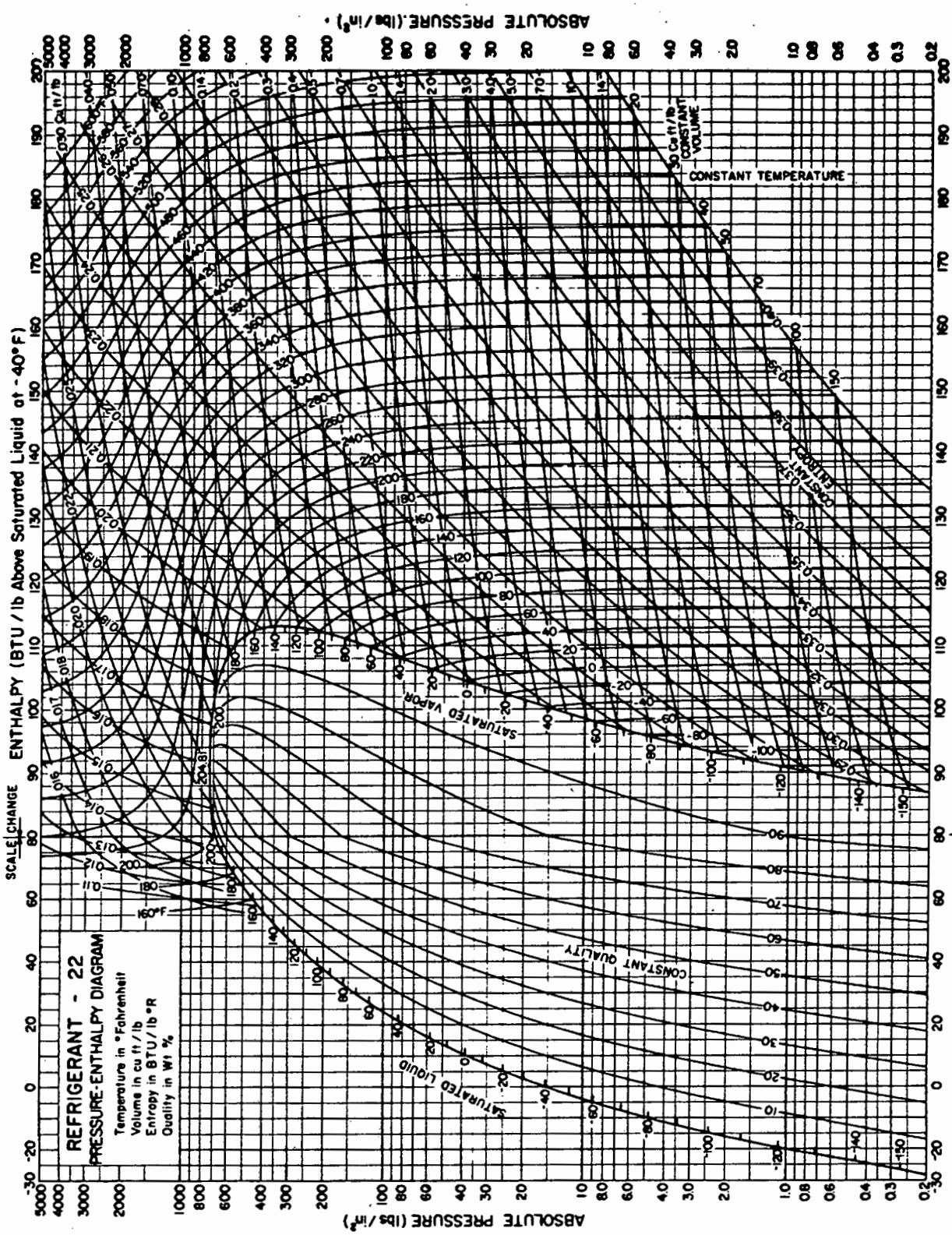
deg C	0	1	2	3	4	5	6	7	8	9	10	deg C
-140	-4.669	-4.694	-4.719	-4.743	-4.768	-4.792	-4.817	-4.841	-4.865	-4.889	-4.912	-140
-130	-4.410	-4.437	-4.463	-4.489	-4.515	-4.541	-4.567	-4.593	-4.618	-4.644	-4.669	-130
-120	-4.138	-4.166	-4.193	-4.221	-4.248	-4.276	-4.303	-4.330	-4.357	-4.384	-4.410	-120
-110	-3.852	-3.881	-3.910	-3.939	-3.968	-3.997	-4.025	-4.053	-4.082	-4.110	-4.138	-110
-100	-3.553	-3.584	-3.614	-3.644	-3.674	-3.704	-3.734	-3.764	-3.793	-3.823	-3.852	-100
-90	-3.242	-3.274	-3.305	-3.337	-3.368	-3.399	-3.430	-3.461	-3.492	-3.523	-3.553	-90
-80	-2.920	-2.953	-2.985	-3.018	-3.050	-3.082	-3.115	-3.147	-3.179	-3.211	-3.242	-80
-70	-2.586	-2.620	-2.654	-2.687	-2.721	-2.754	-2.788	-2.821	-2.854	-2.887	-2.920	-70
-60	-2.243	-2.277	-2.312	-2.347	-2.381	-2.416	-2.450	-2.484	-2.518	-2.552	-2.586	-60
-50	-1.889	-1.925	-1.961	-1.996	-2.032	-2.067	-2.102	-2.137	-2.173	-2.208	-2.243	-50
-40	-1.527	-1.563	-1.600	-1.636	-1.673	-1.709	-1.745	-1.781	-1.817	-1.853	-1.889	-40
-30	-1.156	-1.193	-1.231	-1.268	-1.305	-1.342	-1.379	-1.416	-1.453	-1.490	-1.527	-30
-20	-0.777	-0.816	-0.854	-0.892	-0.930	-0.968	-1.005	-1.043	-1.081	-1.118	-1.156	-20
-10	-0.392	-0.431	-0.469	-0.508	-0.547	-0.585	-0.624	-0.662	-0.701	-0.739	-0.777	-10
- 0	0.000	-0.039	-0.079	-0.118	-0.157	-0.197	-0.236	-0.275	-0.314	-0.353	-0.392	- 0
0	0.000	0.039	0.079	0.119	0.158	0.198	0.238	0.277	0.317	0.357	0.397	0
10	0.397	0.437	0.477	0.517	0.557	0.597	0.637	0.677	0.718	0.758	0.798	10
20	0.798	0.838	0.879	0.919	0.960	1.000	1.041	1.081	1.122	1.162	1.203	20
30	1.203	1.244	1.285	1.325	1.366	1.407	1.448	1.489	1.529	1.570	1.611	30
40	1.611	1.652	1.693	1.734	1.776	1.817	1.858	1.899	1.940	1.981	2.022	40
50	2.022	2.064	2.105	2.146	2.188	2.229	2.270	2.312	2.353	2.394	2.436	50
60	2.436	2.477	2.519	2.560	2.601	2.643	2.684	2.726	2.767	2.809	2.850	60
70	2.850	2.892	2.933	2.975	3.016	3.058	3.100	3.141	3.183	3.224	3.266	70
80	3.266	3.307	3.349	3.390	3.432	3.473	3.515	3.556	3.598	3.639	3.681	80
90	3.681	3.722	3.764	3.805	3.847	3.888	3.930	3.971	4.012	4.054	4.095	90
100	4.095	4.137	4.178	4.219	4.261	4.302	4.343	4.384	4.426	4.467	4.508	100
110	4.508	4.549	4.590	4.632	4.673	4.714	4.755	4.796	4.837	4.878	4.919	110
120	4.919	4.960	5.001	5.042	5.083	5.124	5.164	5.205	5.246	5.287	5.327	120
130	5.327	5.368	5.409	5.450	5.490	5.531	5.571	5.612	5.652	5.693	5.733	130
140	5.733	5.774	5.814	5.855	5.895	5.936	5.976	6.016	6.057	6.097	6.137	140
150	6.137	6.177	6.218	6.258	6.298	6.338	6.378	6.419	6.459	6.499	6.539	150
160	6.539	6.579	6.619	6.659	6.699	6.739	6.779	6.819	6.859	6.899	6.939	160
170	6.939	6.979	7.019	7.059	7.099	7.139	7.179	7.219	7.259	7.299	7.338	170
180	7.338	7.378	7.418	7.458	7.498	7.538	7.578	7.618	7.658	7.697	7.737	180
190	7.737	7.777	7.817	7.857	7.897	7.937	7.977	8.017	8.057	8.097	8.137	190
200	8.137	8.177	8.216	8.256	8.296	8.336	8.376	8.416	8.456	8.497	8.537	200
210	8.537	8.577	8.617	8.657	8.697	8.737	8.777	8.817	8.857	8.898	8.938	210
220	8.938	8.978	9.018	9.058	9.099	9.139	9.179	9.220	9.260	9.300	9.341	220
230	9.341	9.381	9.421	9.462	9.502	9.543	9.583	9.624	9.664	9.705	9.745	230
240	9.745	9.786	9.826	9.867	9.907	9.948	9.989	10.029	10.070	10.111	10.151	240
250	10.151	10.192	10.233	10.274	10.315	10.355	10.396	10.437	10.478	10.519	10.560	250
260	10.560	10.600	10.641	10.682	10.723	10.764	10.805	10.846	10.887	10.928	10.969	260
270	10.969	11.010	11.051	11.093	11.134	11.175	11.216	11.257	11.298	11.339	11.381	270
280	11.381	11.422	11.463	11.504	11.546	11.587	11.628	11.669	11.711	11.752	11.793	280
290	11.793	11.835	11.876	11.918	11.959	12.000	12.042	12.083	12.125	12.166	12.207	290
300	12.207	12.249	12.290	12.332	12.373	12.415	12.456	12.498	12.539	12.581	12.623	300
310	12.623	12.664	12.706	12.747	12.789	12.831	12.872	12.914	12.955	12.997	13.039	310
320	13.039	13.080	13.122	13.164	13.205	13.247	13.289	13.331	13.372	13.414	13.456	320
330	13.456	13.497	13.539	13.581	13.623	13.665	13.706	13.748	13.790	13.832	13.874	330
340	13.874	13.915	13.957	13.999	14.041	14.083	14.125	14.167	14.208	14.250	14.292	340

Absolute thermocouple e.m.f. in Millivolts

Code

K

TC LTD PO BOX 130, CHESTERFIELD, ENGLAND
Telephone 0246 453315
Telex 547748 PROBE G


REFRIGERANT - 22
PRESSURE-ENTHALPY DIAGRAM
Temperature in °Fahrenheit
Volume in cu ft / lb
Entropy in BTU / lb °R
Quality in Wt %
ABSOLUTE PRESSURE (lbs / in²)
ENTHALPY (BTU / lb Above Saturated Liquid at -40°F)
SCALE CHANGE
CONSTANT TEMPERATURE
CONSTANT VOLUME
CONSTANT ENTROPY
CONSTANT QUALITY
SATURATED VAPOR
SATURATED LIQUID

Refrigerant 22 (Chlorodifluoromethane) Properties of Liquid and Saturated Vapor[a]

Temp F	Pressure		Volume cu ft/lb	Density lb/cu ft	Enthalpy[b] Btu/lb		Entropy[b] Btu/(lb) (°R)	
	psia	psig	Vapor v_g	Liquid $1/v_f$	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
−150	0.27163	29.36816[c]	141.23	98.236	−25.974	87.521	−0.07147	0.29501
−145	0.34999	29.20861[c]	111.34	97.800	−24.851	88.100	−0.06787	0.29106
−140	0.44692	29.01126[c]	88.532	97.363	−23.725	88.681	−0.06432	0.28729
−135	0.56584	28.76914[c]	70.981	96.922	−22.596	89.263	−0.06082	0.28369
−130	0.71060	28.47441[c]	57.356	96.480	−21.463	89.848	−0.05736	0.28027
−125	0.88551	28.11829[c]	46.692	96.035	−20.326	90.433	−0.05394	0.27700
−120	1.0954	27.6910[c]	38.280	95.587	−19.185	91.020	−0.05055	0.27388
−115	1.3455	27.1818[c]	31.594	95.137	−18.038	91.608	−0.04720	0.27090
−110	1.6417	26.5788[c]	26.242	94.684	−16.886	92.196	−0.04389	0.26805
−105	1.9903	25.8689[c]	21.930	94.228	−15.729	92.783	−0.04060	0.26533
−100	2.3983	25.0383[c]	18.433	93.770	−14.564	93.371	−0.03734	0.26274
− 98	2.5798	24.6688[c]	17.222	93.585	−14.097	93.606	−0.03605	0.26173
− 96	2.7724	24.2765[c]	16.104	93.401	−13.628	93.840	−0.03476	0.26074
− 94	2.9768	23.8604[c]	15.072	93.215	−13.158	94.075	−0.03347	0.25977
− 92	3.1934	23.4193[c]	14.118	93.030	−12.688	94.309	−0.03219	0.25881
− 90	3.4229	22.9522[c]	13.235	92.843	−12.216	94.544	−0.03091	0.25787
− 88	3.6657	22.4579[c]	12.417	92.657	−11.743	94.777	−0.02963	0.25695
− 86	3.9224	21.9352[c]	11.659	92.469	−11.268	95.011	−0.02836	0.25604
− 84	4.1936	21.3829[c]	10.955	92.282	−10.793	95.244	−0.02709	0.25515
− 82	4.4800	20.7998[c]	10.302	92.093	−10.316	95.478	−0.02583	0.25428
− 80	4.7822	20.1846[c]	9.6949	91.905	− 9.838	95.710	−0.02457	0.25342
− 78	5.1007	19.5361[c]	9.1301	91.715	− 9.359	95.943	−0.02331	0.25257
− 76	5.4363	18.8528[c]	8.6043	91.525	− 8.878	96.175	−0.02206	0.25174
− 74	5.7896	18.1334[c]	8.1145	91.335	− 8.397	96.406	−0.02081	0.25092
− 72	6.1614	17.3766[c]	7.6579	91.144	− 7.914	96.637	−0.01956	0.25012
− 70	6.5522	16.5809[c]	7.2318	90.952	− 7.429	96.868	−0.01832	0.24932
− 68	6.9628	15.7449[c]	6.8339	90.760	− 6.944	97.098	−0.01708	0.24855
− 66	7.3939	14.8671[c]	6.4621	90.568	− 6.457	97.328	−0.01584	0.24778
− 64	7.8463	13.9460[c]	6.1144	90.374	− 5.968	97.557	−0.01460	0.24703
− 62	8.3208	12.9800[c]	5.7891	90.180	− 5.479	97.786	−0.01337	0.24629
− 60	8.8180	11.9677[c]	5.4844	89.986	− 4.987	98.014	−0.01214	0.24556
− 58	9.3388	10.9074[c]	5.1989	89.791	− 4.495	98.241	−0.01092	0.24484
− 56	9.8839	9.7975[c]	4.9312	89.595	− 4.001	98.468	−0.00969	0.24414
− 54	10.454	8.636[c]	4.6799	89.399	− 3.506	98.694	−0.00847	0.24345
− 52	11.051	7.422[c]	4.4440	89.202	− 3.009	98.920	−0.00725	0.24276
− 50	11.674	6.154[c]	4.2224	89.004	− 2.511	99.144	−0.00604	0.24209
− 48	12.324	4.829[c]	4.0149	88.806	− 2.012	99.369	−0.00483	0.24143
− 46	13.004	3.445[c]	3.8179	88.607	− 1.511	99.592	−0.00361	0.24078
− 44	13.712	2.002[c]	3.6334	88.407	− 1.009	99.814	−0.00241	0.24014
− 42	14.451	0.498[c]	3.4596	88.207	− 0.505	100.036	−0.00120	0.23951
− 40	15.222	0.526	3.2957	88.006	0.000	100.257	0.00000	0.23888
− 38	16.024	1.328	3.1412	87.805	0.506	100.477	0.00120	0.23827
− 36	16.859	2.163	2.9954	87.602	1.014	100.696	0.00240	0.23767
− 34	17.728	3.032	2.8578	87.399	1.524	100.914	0.00359	0.23707
− 32	18.633	3.937	2.7278	87.195	2.035	101.132	0.00479	0.23649
− 30	19.573	4.877	3.6049	86.991	2.547	101.348	0.00598	0.23591
− 28	20.549	5.853	2.4887	86.785	3.061	101.564	0.00716	0.23534
− 26	21.564	6.868	2.3787	86.579	3.576	101.778	0.00835	0.23478
− 24	22.617	7.921	2.2746	86.372	4.093	101.992	0.00953	0.23423
− 22	23.711	9.015	2.1760	86.165	4.611	102.204	0.01072	0.23369
− 20	24.845	10.149	2.0826	85.956	5.131	102.415	0.01189	0.23315
− 18	26.020	11.324	1.9940	85.747	5.652	102.626	0.01307	0.23262
− 16	27.239	12.543	1.9099	85.537	6.175	102.835	0.01425	0.23210
− 14	28.501	13.805	1.8302	85.326	6.699	103.043	0.01542	0.23159
− 12	29.809	15.113	1.7544	85.114	7.224	103.250	0.01659	0.23108
− 10	31.162	16.466	1.6825	84.901	7.751	103.455	0.01776	0.23058
− 8	32.563	17.867	1.6141	84.688	8.280	103.660	0.01892	0.23008
− 6	34.011	19.315	1.5491	84.473	8.810	103.863	0.02009	0.22960
− 4	35.509	20.813	1.4872	84.258	9.341	104.065	0.02125	0.22912
− 2	37.057	22.361	1.4283	84.042	9.874	104.266	0.00241	0.22864
0	38.657	23.961	1.3723	83.825	10.409	104.465	0.02357	0.22817
2	40.309	25.613	1.3189	83.606	10.945	104.663	0.02472	0.22771
4	42.014	27.318	1.2680	83.387	11.483	104.860	0.02587	0.22725
5†	42.888	28.192	1.2434	83.277	11.752	104.958	0.02645	0.22703
6	43.775	29.079	1.2195	83.167	12.022	105.056	0.02703	0.22680
8	45.591	30.895	1.1732	82.946	12.562	105.250	0.02818	0.22636
10	47.464	32.768	1.1290	82.724	13.104	105.442	0.02932	0.22592
12	49.396	34.700	1.0869	82.501	13.648	105.633	0.03047	0.22548
14	51.387	36.691	1.0466	82.276	14.193	105.823	0.03161	0.22505
16	53.438	38.742	1.0082	82.051	14.739	106.011	0.03275	0.22463
18	55.551	40.855	0.97144	81.825	15.288	106.198	0.03389	0.22421
20	57.727	43.031	0.93631	81.597	15.837	106.383	0.03503	0.22379
22	59.967	45.271	0.90270	81.368	16.389	106.566	0.03617	0.22338
24	62.272	47.576	0.87055	81.138	16.942	106.748	0.03730	0.22297
26	64.644	49.948	0.83978	80.907	17.496	106.928	0.03844	0.22257
28	67.083	52.387	0.81031	80.675	18.052	107.107	0.03958	0.22217
30	69.591	54.895	0.78208	80.441	18.609	107.284	0.04070	0.22178
32	72.169	57.473	0.75503	80.207	19.169	107.459	0.04182	0.22139
34	74.818	60.122	0.72911	79.971	19.729	107.632	0.04295	0.22100
36	77.540	62.844	0.70425	79.733	20.292	107.804	0.04407	0.22062
38	80.336	65.640	0.68041	79.495	20.856	107.974	0.04520	0.22024
40	83.206	68.510	0.65753	79.255	21.422	108.142	0.04632	0.21986
42	86.153	71.457	0.63557	79.013	21.989	108.308	0.04744	0.21949

Temp F	Pressure		Volume cu ft/lb	Density lb/cu ft	Enthalpy[b] Btu/lb		Entropy[b] Btu/(lb) (°R)	
	psia	psig	Vapor v_g	Liquid $1/v_f$	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
44	89.177	74.481	0.61448	78.770	22.558	108.472	0.04855	0.21912
46	92.280	77.584	0.59422	78.526	23.120	108.634	0.04967	0.21876
48	95.463	80.767	0.57476	78.280	23.701	108.795	0.05079	0.21839
50	98.727	84.031	0.55606	78.033	24.275	108.953	0.05190	0.21803
52	102.07	87.38	0.53808	77.784	24.851	109.109	0.05301	0.21768
54	105.50	90.81	0.52078	77.534	25.429	109.263	0.05412	0.21732
56	109.02	94.32	0.50414	77.282	26.008	109.415	0.05523	0.21697
58	112.62	97.93	0.48813	77.028	26.589	109.564	0.05634	0.21662
60	116.31	101.62	0.47272	76.773	27.172	109.712	0.05745	0.21627
62	120.09	105.39	0.45788	76.515	27.757	109.857	0.05855	0.21592
64	123.96	109.26	0.44358	76.257	28.344	110.000	0.05966	0.21558
66	127.92	113.22	0.42981	75.996	28.932	110.140	0.06076	0.21524
68	131.97	117.28	0.41653	75.733	29.523	110.278	0.06186	0.21490
70	136.12	121.43	0.40373	75.469	30.116	110.414	0.06296	0.21456
72	140.37	125.67	0.39139	75.202	30.710	110.547	0.06406	0.21422
74	144.71	130.01	0.37949	74.934	31.307	110.677	0.06516	0.21388
76	149.15	134.45	0.36800	74.664	31.906	110.805	0.06626	0.21355
78	153.69	138.99	0.35691	74.391	32.506	110.930	0.06736	0.21321
80	158.33	143.63	0.34621	74.116	33.109	111.052	0.06846	0.21288
82	163.07	148.37	0.33587	73.839	33.714	111.171	0.06956	0.21255
84	167.92	153.22	0.32588	73.560	34.322	111.288	0.07065	0.21222
86†	172.87	158.17	0.31623	73.278	34.931	111.401	0.07175	0.21188
88	177.93	163.23	0.30690	72.994	35.543	111.512	0.07285	0.21155
90	183.09	168.40	0.29789	72.708	36.158	111.619	0.07394	0.21122
92	188.37	173.67	0.28917	72.419	36.774	111.723	0.07504	0.21089
94	193.76	179.06	0.28073	72.127	37.394	111.824	0.07613	0.21056
96	199.26	184.56	0.27257	71.833	38.016	111.921	0.07723	0.21023
98	204.87	190.18	0.26467	71.536	38.640	112.015	0.07832	0.20989
100	210.60	195.91	0.25702	71.236	39.267	112.105	0.07942	0.20956
102	216.45	201.76	0.24962	70.933	39.897	112.192	0.08052	0.20923
104	222.42	207.72	0.24244	70.626	40.530	112.274	0.08161	0.20889
106	228.50	213.81	0.23549	70.317	41.166	112.353	0.08271	0.20855
108	234.71	220.02	0.22875	70.005	41.804	112.427	0.08381	0.20821
110	241.04	226.35	0.22222	69.689	42.446	112.498	0.08491	0.20787
112	247.50	232.80	0.21589	69.369	43.091	112.564	0.08601	0.20753
114	254.08	239.38	0.20974	69.046	43.739	112.626	0.08711	0.20718
116	260.79	246.10	0.20378	68.719	44.391	112.682	0.08821	0.20684
118	267.63	252.94	0.19800	68.388	45.046	112.735	0.08932	0.20649
120	274.60	259.91	0.19238	68.054	45.705	112.782	0.09042	0.20613
122	281.71	267.01	0.18692	67.714	46.368	112.824	0.09153	0.20578
124	288.95	274.25	0.18163	67.371	47.034	112.860	0.09264	0.20542
126	296.33	281.63	0.17648	67.023	47.705	112.891	0.09375	0.20505
128	303.84	289.14	0.17147	66.670	48.380	112.917	0.09487	0.20468
130	311.50	296.80	0.16661	66.312	49.059	112.936	0.09598	0.20431
132	319.29	304.60	0.16187	65.949	49.743	112.949	0.09711	0.20393
134	327.23	312.54	0.15727	65.581	50.432	112.955	0.09823	0.20354
136	335.32	320.63	0.15279	65.207	51.125	112.954	0.09936	0.20315
138	343.56	328.86	0.14843	64.826	51.824	112.947	0.10049	0.20275
140	351.94	337.25	0.14418	64.440	52.528	112.931	0.10163	0.20235
142	360.48	345.79	0.14004	64.047	53.238	112.908	0.10277	0.20194
144	369.17	354.48	0.13600	63.647	53.955	112.877	0.10391	0.20152
146	378.02	363.32	0.13207	63.240	54.677	112.836	0.10507	0.20109
148	387.03	372.33	0.12823	62.825	55.406	112.787	0.10622	0.20065
150	369.19	381.50	0.12448	62.402	56.143	112.728	0.10739	0.20020
152	405.52	390.83	0.12083	61.970	56.887	112.658	0.10856	0.19974
154	415.02	400.32	0.11726	61.529	57.638	112.577	0.10974	0.19926
156	424.68	409.99	0.11376	61.079	58.399	112.485	0.11093	0.19873
158	434.52	419.82	0.11035	60.617	59.168	112.381	0.11213	0.19823
160	444.53	429.83	0.10701	60.145	59.948	112.263	0.11334	0.19776
162	454.71	440.01	0.10374	59.660	60.737	112.131	0.11456	0.19723
164	465.07	450.37	0.10054	59.163	61.538	111.984	0.11580	0.19668
166	475.61	460.92	0.097393	58.651	62.351	111.820	0.11705	0.19611
168	486.34	471.65	0.094309	58.125	63.178	111.639	0.11831	0.19552
170	497.26	482.56	0.091279	57.581	64.019	111.438	0.11959	0.19490
172	508.37	493.67	0.088299	57.019	64.875	111.216	0.12089	0.19425
174	519.67	504.97	0.085365	56.438	65.750	110.970	0.12222	0.19358
176	531.17	516.47	0.082473	55.834	66.643	110.699	0.12356	0.19287
178	542.87	528.18	0.079616	55.205	67.558	110.400	0.12494	0.19212
180	554.78	540.09	0.076790	54.549	68.498	110.068	0.12635	0.19133
182	566.90	552.21	0.073987	53.861	69.465	109.700	0.12779	0.19050
184	579.24	564.54	0.071201	53.136	70.464	109.290	0.12928	0.18960
186	591.80	577.10	0.068421	52.370	71.500	108.832	0.13082	0.18864
188	604.58	589.88	0.065638	51.553	72.579	108.317	0.13242	0.18760
190	617.59	602.89	0.062837	50.677	73.711	107.734	0.13409	0.18646
192	630.84	616.14	0.059999	49.728	74.907	107.067	0.13585	0.18520
194	644.33	629.64	0.057006	48.685	76.184	106.295	0.13773	0.18380
196	658.08	643.38	0.054089	47.518	77.568	105.381	0.13977	0.18218
198	672.08	657.39	0.050912	46.178	79.102	104.270	0.14202	0.18029
200	686.36	671.66	0.047438	44.571	80.862	102.853	0.14460	0.17794
202	700.91	686.21	0.043375	42.476	83.030	100.870	0.14779	0.17475
204	715.75	701.05	0.037545	38.991	86.309	97.260	0.15264	0.16914
204.81	721.91	707.21	0.030525	32.760	91.329	91.329	0.16016	0.16016

[a] From published data (1964) of E. I. du Pont de Nemours & Co., Inc. Used by permission.
[b] Inches of mercury below one standard atmosphere.
[c] Based on 0 for the saturated liquid at −40 F.
[d] Standard cycle temperatures.

PHYSICAL PROPERTIES
** OF DIMETHYL ETHER - TETRA ETHYLENE GLYCOL **

Chemical Name : Dimethyl ether tetra ethylene glycol
Also
Dimethoxy tetra glycol
bis (2 - methoxyethoxyethyl ether)

Chemical Formula: $CH_3 (O CH_2 CH_2)_4 OCH_3$

Common Name : Tetraglyme

Containers : Drums

Uses : Solvent

Physical Properties

Water white practically odourless liquid;
Stable; Soluble in hydrocarbons and water;
Specific gravity = 1,0132 at 20 ^{o}C
Boiling point = 275,8 ^{o}C at 760 mm Hg
= 189,0 ^{o}C at 100 mm Hg
Freezing point = - 29,7 ^{o}C
Vapour pressure < 0,01 mm Hg at 20 ^{o}C
Flash point = 285 ^{o}C
Viscosity = 0,0405 poise (20 ^{o}C)

Source: Merck Index